MECHANICAL DRAFT.

STURTEVANT

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MECHANICAL

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A PRACTICAL TREATISE

B. F. STURTEVANT CO.
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STURTEVANT ENGINEERING CO.
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Introduction.

The subject of mechanical draft has been discussed at greater or less length in the technical press and before various engineering societies; but in all cases such discussion has been distinctly limited. Here, for the first time, the attempt is made to give the treatment that its importance demands. Although its introduction is evidence of a somewhat radical departure in certain features of boiler practice, yet extended and recent experience clearly indicates the permanence of this departure. Hereafter, the progressive engineer will not adopt the chimney as the sole means of draft production, but will first carefully consider the respective advantages of the chimney and the fan, for the claims of the latter to superiority cannot be disregarded.

Manifestly, the individual who in the present day is to decide between the two methods, requires a clear presentation of their respective merits, such as is attempted in this work. Its object is two-fold: First, to instruct by a lucid discussion of the entire subject, with such supplementary information as may be necessary to show the superiority of mechanical draft. Second, to show the special adaptability of the Sturtevant fans for this purpose, and to indicate in some degree the extent to which they have already been applied.

With the desire to give the reader all the information that may be required for a full understanding of the subject, without necessitating reference to other works, the chapters on Water, Steam and Fuels have been incorporated. In the chapters on Efficiency of Fuels, and of Boilers, the principles which pertain to the efficiency, convenience and adaptability of mechanical draft are presented in their abstract relation to the subject in general. In so far as such statements are general in their character their treatment is that of a text-book. But wherever they particularly concern the utility of mechanical draft and the employment of fans for its production, they are substantiated by quotations from eminent authorities, and in all cases references are introduced. An impartial character is thus given to all statements, while the actual facts are doubly emphasized.

The last two chapters are of especial value and interest, inasmuch as they forcibly indicate the methods of application under a great variety of conditions, and are witnesses to the possibilities of this method of draft production. In so far as possible, all statements regarding the application and operation of Sturtevant fans, as well as those relating to the character and advantages of certain other systems or devices, have been quoted and the references given. This fact should be carefully noted, for the purpose has been to entirely eliminate the element of personal prejudice and thereby avoid even the appearance of either favorable or adverse comment. To the reader who can give but little time to this discussion, Chapter XI serves as a summary of the advantages of this system.

Every endeavor has been made to render this work authoritative, to discuss the subject in all its aspects and to give assurance of the most impartial treatment. In the light of the foregoing it is presented in confidence that its careful perusal will result in conviction as to the superiority of mechanical draft.

B. F. STURTEVANT CO.

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CHAPTER I.

WATER.

Composition. — Pure water, whether solid, liquid or gaseous, is a chemical combination of the two elements hydrogen and oxygen in the unalterable proportion of two parts, by volume, of the former to one part of the latter. A mere mechanical mixture of hydrogen and oxygen remains still a mixture until, through the influence of heat, electricity or other special agents, the two chemically combine. Such a result may be brought about by introducing a lighted taper in a properly proportioned mixture of the gases, the resulting water being deposited as dew upon the sides of the containing vessel. If the union of the hydrogen and oxygen be effected in an apparatus so arranged that the water formed by the combination is kept at such a high temperature that it remains in the gaseous condition, it will be found that the two volumes of hydrogen and one volume of oxygen which were mixed together have become compacted into two volumes of steam, as the result of the chemical union.

Conversely, two volumes of water in its vaporized or gaseous form may, by various methods, be decomposed into its constituent elements; namely, two volumes of hydrogen and one volume of oxygen. Principal among the methods of dissociation is the application of heat. Consequently the presence of moisture in fuel may assume considerable importance in the ordinary process of combustion in connection with steam boilers.

Weight and Bulk. — Water is universally adopted as the standard by which the relative weights of all liquids and solids are determined, this relation being expressed by the term "specific gravity." The specific gravity of a body, therefore, indicates its weight as compared with that of an equal volume of pure water. Determinations of specific gravity are generally referred to the weight of one cubic foot of water at 62° F. At the more important temperatures the weights are as follows: —

Weight of One Cubic Foot of Pure Water.

At 32° F.	(freezing point)	62.418	pounds.
" 39.1° F.	(maximum density)	62.425	"
" 62° F.	(standard temperature)	62.355	"
" 212° F.	(boiling point under atmospheric pressure)	59.640	"

For general purposes the weight of water is taken in round numbers as 62.5 pounds per cubic foot. The calculated weights at temperatures from 32° to 290° are given in Table No. 1.

In bulk, water is usually measured by the gallon, the volume of which is 231 cubic inches (the British gallon contains 277.274 cubic inches), or 0.134 cubic feet. A gallon of water at 62°, therefore, weighs slightly over 8⅓ pounds, and 7.48 gallons equal one cubic foot.

Table No. 1. — Expansion and Density of Pure Water.

Temperature. Degrees Fahr.	Absolute Pressure of Vapor per sq. in. Pounds.	Relative Volume. Water at 32° = 1.	Relative Density. Water at 32° = 1.	Density or Weight of 1 Cubic Foot. Pounds.	Temperature. Degrees Fahr.	Absolute Pressure of Vapor per sq. in. Pounds.	Relative Volume. Water at 32° = 1.	Relative Density. Water at 32° = 1.	Density or Weight of 1 Cubic Foot. Pounds.
32°	0.089	1.00000	1.00000	62.418	135°	2.542	1.01539	0.98484	61.472
35	0.100	0.99993	1.00007	62.422	140	2.879	1.01690	0.98339	61.381
40	0.122	0.99989	1.00011	62.425	145	3.273	1.01839	0.98194	61.291
45	0.147	0.99993	1.00007	62.422	150	3.708	1.01989	0.98050	61.201
50	0.178	1.00015	0.99985	62.409	155	4.193	1.02164	0.97882	61.096
55	0.214	1.00038	0.99961	62.394	160	4.731	1.02340	0.97714	60.991
60	0.254	1.00074	0.99926	62.372	165	5.327	1.02589	0.97477	60.843
65	0.304	1.00119	0.99881	62.344	170	5.985	1.02690	0.97380	60.783
70	0.360	1.00160	0.99832	62.313	175	6.708	1.02906	0.97193	60.665
75	0.427	1.00239	0.99771	62.275	180	7.511	1.03100	0.97006	60.548
80	0.503	1.00299	0.99702	62.232	185	8.375	1.03300	0.96828	60.430
85	0.592	1.00379	0.99622	62.182	190	9.335	1.03500	0.96632	60.314
90	0.693	1.00459	0.99543	62.133	195	10.385	1.03700	0.96440	60.198
95	0.809	1.00554	0.99449	62.074	200	11.526	1.03889	0.96256	60.081
100	0.940	1.00639	0.99365	62.022	205	12.770	1.0414	0.9602	59.93
105	1.095	1.00739	0.99260	61.960	210	14.126	1.0434	0.9584	59.82
110	1.267	1.00889	0.99119	61.868	212	14.7	1.0444	0.9575	59.76
115	1.462	1.00989	0.99021	61.807	230	20.87	1.0529	0.9499	59.36
120	1.685	1.01139	0.98874	61.715	250	29.80	1.0628	0.9411	58.75
125	1.932	1.01239	0.98808	61.654	270	41.87	1.0727	0.9323	58.18
130	2.215	1.01390	0.98630	61.563	290	57.64	1.0838	0.9227	57.59

Expansion by Heat. — Although water is practically non-compressible, even under the most extraordinary pressure, it readily expands by the mere application of heat, with the notable exception that between the temperature of melting ice at 32° and that of greatest density at 39.1° there is a gradual contraction in volume as heat is applied.

The rate of expansion is, however, variable, and above 212° but little is known regarding it experimentally. An application of the formula derived from experiments made below 212° gives at least approximate values. In Table No. 1 are embodied the results of calculation by Rankine's approximate formula, which gives substantially correct values at low and moderate temperatures. The results at high temperatures are, however, too large, as is evidenced by the fact that by the table the density at 212° is shown to be 59.76 pounds per cubic foot, while by actual measurement it has been found to be 59.64 pounds,—an error, however, of only 0.002 in the value, which is of small account except in refined calculations.

Specific Heat.—Bodies vary greatly in the capacity which they possess for absorbing heat under equal changes in temperature. The relation which thus exists between them is expressed by the "specific heat," which may be defined as the quantity of heat necessary to be imparted to a given body in order to raise its temperature one degree relatively to the quantity that is required to raise through one degree an equal weight of water from its point of greatest density at 39.1°. Thus, for instance, one pound of air at constant pressure may be raised through one degree by the expenditure of only 0.2375 of the heat necessary to raise one pound of water through one degree; or, what amounts to the same, the amount of heat expended to raise the temperature of one pound of water by one degree would heat $\frac{1}{0.2375} = 4.2105$ pounds of air through the same increment. As the specific heat of water is greater than that of any other known substance, the specific heat of all other substances must of necessity be expressed in decimals. Water does not absorb heat exactly in proportion to its increase in temperature; in other words, the specific heat of water varies with the temperature, as is rendered evident by Table No. 2.

Table No. 2.—Specific Heat of Water.

Temperature. Degrees Fahrenheit.	Specific Heat at the Given Temperature. Freezing point = 1.	Temperature. Degrees Fahrenheit.	Specific Heat at the Given Temperature. Freezing point = 1.	Temperature. Degrees Fahrenheit.	Specific Heat at the Given Temperature. Freezing point = 1.
32°	1.0000	176°	1.0089	320°	1.0294
50	1.0005	194	1.0109	338	1.0328
68	1.0012	212	1.0130	356	1.0364
86	1.0020	230	1.0153	374	1.0401
104	1.0030	248	1.0177	394	1.0440
122	1.0042	266	1.0204	410	1.0481
140	1.0056	284	1.0232	428	1.0524
158	1.0072	302	1.0262	446	1.0568

Unit of Heat.—The quantitative measure of heat is the thermal unit. The British thermal unit (as distinguished from the French thermal unit, or *calorie*) is that quantity of heat which is required to raise the temperature of one pound of pure water through one degree Fahr. at or near 39.1° Fahr., the temperature of maximum density of water. As employed in general practice, the term is usually abbreviated to "B. T. U."

The relation existing between the temperature of water in degrees Fahrenheit and the number of thermal units contained therein, together with the increase in the number of thermal units for each increment of temperature of 5 degrees, is indicated in Table No. 3.

Table No. 3.—Number of Thermal Units Contained in One Pound of Water.

Temperature. Degs. F.	Number of Thermal Units.	Increase.	Temperature. Degs. F.	Number of Thermal Units.	Increase.	Temperature. Degs. F.	Number of Thermal Units.	Increase.
35°	35.000		155°	155.339	5.034	275°	276.985	5.107
40	40.001	5.001	160	160.374	5.035	280	282.095	5.110
45	45.002	5.001	165	165.413	5.039	285	287.210	5.115
50	50.003	5.001	170	170.453	5.040	290	292.329	5.119
55	55.006	5.003	175	175.497	5.044	295	297.452	5.123
60	60.009	5.003	180	180.542	5.045	300	302.580	5.128
65	65.014	5.005	185	185.591	4.049	305	307.712	5.132
70	70.020	5.006	190	190.643	5.052	310	312.848	5.136
75	75.027	5.007	195	195.697	5.054	315	317.988	5.140
80	80.036	5.009	200	200.753	5.056	320	323.134	5.146
85	85.045	5.009	205	205.813	5.060	325	328.284	5.150
90	90.055	5.010	210	210.874	5.061	330	333.438	5.154
95	95.067	5.012	215	215.939	5.065	335	338.596	5.158
100	100.080	5.013	220	221.007	5.068	340	343.759	5.163
105	105.095	5.015	225	226.078	5.071	345	348.927	5.168
110	110.110	5.015	230	231.153	5.075	350	354.101	5.174
115	115.129	5.019	235	236.232	5.079	355	359.280	5.179
120	120.149	5.020	240	241.313	5.081	360	364.464	5.184
125	125.169	5.020	245	246.398	5.085	365	369.653	5.189
130	130.192	5.023	250	251.487	5.089	370	374.846	5.193
135	135.217	5.025	255	256.579	5.092	375	380.044	5.198
140	140.245	5.028	260	261.674	5.095	380	385.247	3.203
145	145.175	5.030	265	266.774	5.100	385	390.456	5.209
150	150.305	5.030	270	271.878	5.104	390	395.672	5.216

Mechanical Equivalent of Heat. — The mechanical unit of work is the "foot-pound," or the work required to raise one pound through a distance of one foot. The mechanical theory of heat regards heat as a mode of motion, and investigation has shown that there exists a definite relation between these two forms of energy, which is known as the "mechanical equivalent of heat." That is, if, as in the experiments of Joule, a certain known amount of mechanical energy is expended (as by the falling of a weight) to operate paddles in a vessel of water, the increase in temperature of the water, due to agitation by the paddles, will always be found to be proportional to the work done. This relation or proportion is universally expressed by the amount of work necessary to raise the temperature of one pound of water through one degree Fahrenheit, and the latest experimental determinations of Rowland show it to be practically 778 foot-pounds.

Pressure of Water. — From the weight of water at the standard temperature of 62°, its pressure upon any exposed surface may be readily determined for any given depth or head. The weight of one cubic foot at the above temperature being 62.355 pounds, it is evident that for a head of one foot the pressure must be 62.355 pounds per square foot, and $\frac{62.355}{144} = 0.433$ pounds per square inch ; and, further, that a pressure of one pound per square inch will be produced by a head of $\frac{1}{0.433} = 2.309$ feet. Upon this basis Table No. 4 has been calculated to show the head corresponding to different pressures.

Table No. 4. — Head in Feet of Water Corresponding to Pressures in Pounds per Square Inch.

Pressure.	0	1	2	3	4	5	6	7	8	9
0		2.309	4.619	6.928	9.238	11.547	13.857	16.166	18.476	20.785
10	23.095	25.404	27.714	30.023	32.333	34.642	36.952	39.261	41.570	43.880
20	46.189	48.499	50.808	53.118	55.427	57.737	60.046	62.356	64.665	66.975
30	69.284	71.594	73.903	76.213	78.522	80.831	83.141	85.450	87.760	90.069
40	92.379	94.688	96.998	99.307	101.62	103.93	106.24	108.55	110.85	113.16
50	115.474	117.78	120.09	122.40	124.71	126.02	129.33	131.64	133.95	136.26
60	138.568	140.88	143.19	145.50	147.81	150.12	152.42	154.73	157.04	159.35
70	161.663	163.97	166.28	168.59	170.90	173.21	175.52	177.83	180.14	182.45
80	184.758	187.07	189.38	191.69	194.00	196.31	198.61	200.92	203.23	205.54
90	207.852	210.16	212.47	214.78	217.09	219.40	221.71	224.02	226.33	228.64

Impurities in Water. — Water, as present in nature, is always more or less contaminated by impurities, so that for all refined measurements in which water plays the part of a standard it must be in its distilled state; that is, absolutely pure.

In boiler practice, the exact composition of the water is of marked importance, as upon the impurities held in suspension or solution may depend, not only the economy of operation, but the very life of the boiler itself. The more common impurities, the character of the trouble which they cause and the remedy or palliation to be applied, are indicated in the following list: —

Troublesome Substance.	*Trouble.*	*Remedy or Palliation.*
Sediment, mud, clay, etc.	Incrustation.	Filtration. Blowing off.
Readily soluble salts.	Incrustation.	Blowing off.
Bicarbonate of lime, magnesia and iron.	Incrustation.	Heating feed, addition of caustic soda, lime or magnesia, etc.
Sulphate of lime.	Incrustation.	Addition of carbonate of soda, barium chloride.
Chloride or sulphate of magnesium.	Corrosion.	Addition of carbonate of soda, etc.
Carbonate of soda in large amounts.	Priming.	Addition of barium chloride, etc.
Acid (in mine waters).	Corrosion.	Alkali.
Dissolved carbonic acid and oxygen.	Corrosion.	Heating feed, addition of caustic soda, slaked lime, etc.
Grease (from condensed steam).	Corrosion.	Slaked lime and filtering, carbonate of soda, substitute mineral oil.
Organic matter (sewage).	Priming.	Precipitate with alum or ferric chloride and filter.
Organic matter.	Corrosion.	Precipitate with alum or ferric chloride and filter.

CHAPTER II.

STEAM.

Composition. — Water in its aëriform condition is to be considered as a vapor rather than as a gas, the term "gas" applying more properly to a body which, under ordinary temperature and pressure, continually remains in its aëriform state. In its vaporous condition, water exists in the atmosphere in various proportions and under different conditions of atmospheric pressure, as indicated in Table No. 5. The slow process by which this production of aqueous vapor takes place at the free surface of a liquid is generally termed *evaporation*, while the more rapid production of the vapor throughout the mass is commonly designated as *boiling*. Under either method of vaporization the change is merely physical, the constituent elements remaining the same in character and proportion; namely, two parts of hydrogen and one part of oxygen, by volume.

Weight and Bulk. — When, by the application of heat, aqueous vapor is produced from, and in contact with, water in a closed vessel, it is usually denominated "steam," and under these conditions is always saturated. Saturated steam is of varying density and temperature, according to the pressure under which it is generated; but there exists an unalterable relation between density, elastic force and temperature, such that if one of these properties remains constant the others so remain, while a change in one results in a change of the other two, and always in a fixed ratio.

The elastic force or pressure at low temperatures is shown in column 4 of Table No. 5, while that at higher temperatures, as expressed in pounds per square inch above vacuum, is presented in Table No. 6. In columns 5 and 6 of this table are also given the weights and volumes of steam under the corresponding temperatures and pressures. The specific density of gaseous steam is 0.622, that of air being 1. That is to say, the weight of a cubic foot of gaseous steam is about five-eighths of that of a cubic foot of air of same pressure and temperature.

Expansion by Heat, Absolute Zero. — When saturated steam is superheated or surcharged with heat, as may only be done when it is separated from the water from which it was generated, it advances from the condition of saturation to that of gaseity. The gaseous state is only arrived at by considerably elevating the temperature, supposing the pressure remains the same. Obviously, the test of perfect gaseity must be the uniformity of the rate of expansion with the rise in temperature. Experiment has shown that, with the exception of a slight variation at a temperature just above that at which it was generated, steam, thus superheated, follows the law that controls the expansion of permanent gases: this law being that, the temperature remaining the same, the volume of a given quantity of gas is inversely proportional to the pressure which it sustains; or, conversely, that, the pressure remaining the same, the volume will be proportional to the temperature. Therefore, the density or volume at any given temperature being known, it may be readily determined by proportion for any other temperature.

This rate or coefficient of expansion of a perfect gas per degree, as determined by the most recent and refined experiments, is $0.00203 = \frac{1}{492.66}$ at the freezing point, or 32° F. Or, in other words, for each rise in temperature of one degree, the gas of freezing temperature increases in volume $\frac{1}{492.66}$, and an increase of 492.66° would double the volume. If, then, this law holds good, reckoning upward from freezing point, as has been conclusively proven by experiment, it is reasonable to suppose that it likewise holds good reckoning downward, and that for every degree of temperature withdrawn from the gas it is diminished $\frac{1}{492.66}$ of its volume at 32°. Carried to its limit, this would point to the fact that at 492.66° below freezing (or 460.66° below zero Fahrenheit) the contraction would be equal to the volume; that is, the volume would cease to exist. This is the so-called *absolute zero* of temperature, the starting-point for the scale of absolute temperatures by which the proportional expansion of gases is determined. Without appreciable error it may be expressed in round numbers as 461° below zero Fahrenheit, and will be so understood in calculations which follow.

The action of steam under the imposed conditions of constant pressure, superheating and separation from water is not to be confounded with that which takes places in the ordinary boiler or in the cylinder of a steam engine. In the case of a boiler, an increase in the temperature of the steam cannot occur without an equal increase in that of the water, which results in the generation of more steam and an increase in pressure; in fact, the relations are such that the temperature of the steam may always be determined from a knowledge of the pressure, and *vice versa.*

In an engine cylinder, on the other hand, the steam in the process of expansion expends a certain portion of its energy in work, its pressure and temperature are both reduced, and its volume is increased. But, owing to condensation by loss of heat transformed into work and to local conditions, the rate of expansion under the decreasing pressure departs from the law of a perfect gas.

Table No. 5.— Weights of Air, Vapor of Water, and Saturated Mixtures of Air and Vapor at different Temperatures, under the ordinary Atmospheric Pressure of 29.921 Inches of Mercury.

				Mixtures of Air saturated with Vapor.						
					Weight of Cubic Foot of the Mixture of Air and Vapor.					
Temperature. Degs. Fahrenheit 1	Volume of Dry Air at different Temperatures, the Volume at 32° being 1.000. 2	Weight of a Cubic Foot of Dry Air at different Temperatures, in Pounds. 3	Elastic Force of Vapor in Inches of Mercury, Regnault. 4	Elastic Force of the Air in the Mixture of Air and Vapor in Ins. of Mercury. 5	Weight of the Air, in Pounds. 6	Weight of the Vapor, in Pounds 7	Total Weight of Mixture, in Pounds 8	Weight of Vapor mixed with 1 lb. of Air, in Pounds. 9	Weight of Dry Air mixed with 1 lb. of Vapor, in Pounds 10	Cubic Ft. of Vapor from 1 lb. of Water at its own Pressure in Column 4 11
0°	.935	.0864	.044	29.877	.0863	.000079	.086379	.00092	1092.4	
12	.960	.0842	.074	29.849	.0840	.000130	.084130	.00155	646.1	
22	.980	.0824	.118	29.803	.0821	.000202	.082302	.00245	406.4	
32	1.000	.0807	.181	29.740	.0802	.000304	.080504	.00379	263.81	3,289
42	1.020	.0791	.267	29.654	.0784	.000440	.078840	.00561	178.18	2,252
52	1.041	.0776	.388	29.533	.0766	.000627	.077227	.00819	122.17	1,595
62	1.061	.0761	.556	29.365	.0747	.000881	.075581	.01179	84.79	1,135
72	1.082	.0747	.785	29.136	.0727	.001221	.073921	.01680	59.54	819
82	1.102	.0733	1.092	28.829	.0706	.001667	.072267	.02361	42.35	600
92	1.122	.0720	1.501	28.420	.0684	.002250	.070717	.03289	30.40	444
102	1.143	.0707	2.036	27.885	.0659	.002997	.068897	.04547	21.98	334
112	1.163	.0694	2.731	27.190	.0631	.003946	.067046	.06253	15.99	253
122	1.184	.0682	3.621	26.300	.0599	.005142	.065042	.08584	11.65	194
132	1.204	.0671	4.752	25.169	.0564	.006639	.063039	.11771	8.49	151
142	1.224	.0660	6.165	23.756	.0524	.008473	.060873	.16170	6.18	118
152	1.245	.0649	7.930	21.991	.0477	.010716	.058416	.22465	4.45	93.3
162	1.265	.0638	10.099	19.822	.0423	.013415	.055715	.31713	3.15	74.5
172	1.285	.0628	12.758	17.163	.0360	.016682	.052682	.46338	2.16	59.2
182	1.306	.0618	15.960	13.961	.0288	.020536	.049336	.71300	1.402	48.6
192	1.326	.0609	19.828	10.093	.0205	.025142	.045642	1.22643	.815	39.8
202	1.347	.0600	24.450	5.471	.0109	.030545	.041445	2.80230	.357	32.7
212	1.367	.0591	29.921	0.000	.0000	.036820	.036820	Infinite.	.000	27.1

In fact, a steam engine is a device for transforming heat into work. In so far as all other losses are prevented it becomes a perfect heat engine, performing solely the function of transformation. The indicator diagram serves to show the actual process of expansion as it takes place in the cylinder, subject to practical conditions.

Table No. 6. — Properties of Saturated Steam.

Total Pressure, in lbs. per sq. in., measured from a Vacuum.	Temperature, in Degrees Fahrenheit of Steam and of the Water from which it was evaporated.	Number of British Thermal Units contained in one Pound, reckoned from Zero Fahrenheit.		Weight of one Cubic Foot of Steam, in Decimals of a Pound.	Volume of one Pound of Steam, in Cubic Feet.	Relative Volume or Cubic Feet of Steam from one Cubic Foot of Water.
		Number required for Evaporation, known as Latent Heat, or Heat of Vaporization.	Total Number contained in the Steam.			
1	2	3	4	5	6	7
1	102.	1,042.96	1,145.05	.0030	330.36	20,620.
2	126.27	1,026.01	1,152.45	.0058	172.08	10,720.
3	141.62	1,015.25	1,157.13	.0085	117.52	7,326.
4	153.07	1,007.23	1,160.62	.0112	89.62	5,600.
5	162.33	1,000.73	1,163.45	.0137	72.66	4,535.
6	170.12	995.25	1,165.83	.0163	61.21	3,814.
7	176.91	990.47	1,167.90	.0189	52.94	3,300.
8	182.91	986.25	1,169.73	.0214	46.69	2,910.
9	188.32	982.43	1,171.37	.0239	41.79	2,607.
10	193.24	978.96	1,172.88	.0264	37.84	2,360.
12	201.96	972.80	1,175.54	.0313	31.95	1,988.
14	209.56	967.43	1,177.85	.0362	27.63	1,722.
14.7	212.	965.7	1,178.60	.0380	26.36	1,644.
16	216.30	962.66	1,179.91	.0413	24.21	1,514.
18	222.38	958.34	1,181.76	.0462	21.64	1,350.6
20	227.92	954.41	1,183.45	.0511	19.57	1,220.3
22	233.02	950.79	1,185.01	.0561	17.83	1,113.5
24	237.75	947.42	1,186.45	.0610	16.39	1,024.1
26	242.17	944.28	1,187.80	.0658	15.19	948.4
28	246.33	941.32	1,189.07	.0707	14.14	883.2
30	250.24	938.92	1,190.26	.0755	13.25	826.8
32	253.95	935.88	1,191.39	.0803	12.45	777.2
34	257.48	933.37	1,192.47	.0851	11.75	733.5
36	260.83	930.97	1,193.49	.0899	11.11	694.5
38	264.05	928.67	1,194.47	.0946	10.56	659.7
40	267.12	926.47	1,195.41	.0994	10.06	628.2
42	270.07	924.36	1,196.31	.1041	9.59	599.3
44	272.91	922.32	1,197.18	.1088	9.18	573.7
46	275.65	920.36	1,198.01	.1134	8.82	550.4
48	278.30	918.47	1,198.82	.1181	8.47	529.0
50	280.85	916.63	1,199.60	.1227	8.15	508.5
52	283.33	914.86	1,200.35	.1274	7.85	490.1
54	285.72	913.13	1,201.09	.1320	7.58	472.9
56	288.05	911.46	1,201.80	.1366	7.32	457.0
58	290.32	909.83	1,202.49	.1411	7.08	442.4

Table No. 6. — Properties of Saturated Steam. — Concluded.

1	2	3	4	5	6	7
60	292.52	908.25	1,203.16	.1457	6.86	428.5
62	294.66	906.70	1,203.81	.1502	6.66	415.6
64	296.75	905.20	1,204.45	.1547	6.46	403.5
66	298.79	903.73	1,205.07	.1592	6.28	392.1
68	300.78	902.30	1,205.68	.1637	6.10	381.3
70	302.72	900.90	1,206.27	.1682	5.95	371.2
72	304.62	899.53	1,206.85	.1726	5.80	361.7
74	306.47	898.19	1,207.41	.1770	5.65	352.6
76	308.29	896.88	1,207.97	.1814	5.51	344.1
78	310.07	895.59	1,208.51	.1858	5.34	336.0
80	311.81	894.33	1,209.04	.1901	5.26	328.3
82	313.52	893.09	1,209.56	.1945	5.14	320.9
84	315.19	891.88	1,210.07	.1989	5.03	313.9
86	316.84	890.69	1,210.58	.2032	4.92	307.2
88	318.45	889.52	1,211.07	.2075	4.82	300.8
90	320.04	888.38	1,211.55	.2118	4.72	294.7
92	321.60	887.25	1,212.03	.2161	4.63	288.9
94	323.13	886.14	1,212.49	.2204	4.54	283.3
96	324.63	885.04	1,212.95	.2245	4.44	278.0
98	326.11	883.97	1,213.40	.2288	4.37	272.8
100	327.57	882.91	1,213.85	.2330	4.29	267.9
105	331.11	880.34	1,214.93	.2434	4.11	256.5
110	334.52	877.86	1,215.97	.2538	3.94	246.0
115	337.81	875.47	1,216.97	.2640	3.79	236.3
120	340.99	873.15	1,217.94	.2743	3.65	227.6
125	344.07	870.91	1,218.88	.2843	3.52	219.7
130	347.06	868.73	1,219.79	.2942	3.40	212.3
135	349.95	866.62	1,220.68	.3040	3.29	205.4
140	352.77	864.57	1,221.53	.3139	3.19	199.0
145	355.50	862.57	1,222.37	.3239	3.09	193.0
150	358.16	860.62	1,223.18	.3340	2.99	187.5
160	363.28	856.87	1,224.74	.3521	2.84	177.3
170	368.16	853.29	1,226.23	.3709	2.69	168.4
180	372.82	849.87	1,227.65	.3889	2.57	160.4
190	377.29	846.58	1,229.01	.4072	2.45	153.4
200	381.57	843.43	1,230.32	.4250	2.35	147.1
250	401.07	831.22	1,235.73	.5464	1.83	114.
300	418.22	819.61	1,240.74	.6486	1.54	96.
350	431.96	810.69	1,244.58	.7498	1.33	83.
400	444.92	800.20	1,249.09	.8502	1.18	73.

Specific Heat. — The specific heat of saturated steam is 0.305 referred to water as a standard; that is, it requires only 0.305 as much heat to raise the temperature of a given weight of saturated steam through one degree as would be necessary to raise the same weight of water through the same increment. Properly, this value of 0.305 is the specific heat of the water and its saturated vapor combined. The specific heat of gaseous steam is 0.475.

Latent and Sensible Heat. — When water is heated in the open atmosphere its temperature gradually increases until 212° is reached. Further application of heat has no effect in raising the temperature beyond this point, but the water boils and passes off as steam, the temperature of which will also be found to be 212°. Evidently, then, the heat applied to accomplish the vaporization cannot be measured by the thermometer. But experiment has shown that in the evaporation of the entire volume of water there thus disappears about five and a half times as much heat as is required to raise it from freezing to boiling point. On the other hand, it has been proven by experiment that upon the condensation of steam there is relinquished a large quantity of heat of which the thermometer gave no indication, and that this amount is exactly equal to that which disappeared in the process of vaporization. Because of the hidden character of this heat it is known as *latent heat*, and is measured in thermal units, while that indicated by the thermometer is designated as *sensible heat*. The sum of the heat units in the water and in the steam generated therefrom is known as the *total heat*.

The total heat required in the formation of steam is expended in three ways.

1. In raising the temperature of the water to the boiling point. This would be exactly measured by the sensible heat if the specific heat of water was constant. The total heat of the water in heat units varies slightly, however, from the temperature in degrees as indicated by the thermometer. This relation is clearly shown in Table No. 3.

2. In the work done in transforming the water into steam. This is distinctly an internal work, and consists in separating the water particles and establishing a repulsive action between them.

3. In the additional work of overcoming the incumbent pressure of the surrounding atmosphere so that enlargement of volume may take place. This work is entirely external.

An analysis of the heat and work expenditures in increasing the temperature of one pound of water from 32° to the boiling point, and transforming it into steam of 212° temperature under atmospheric pressure, is displayed in Table No. 7, from which the relative amount of heat expended in each of the three ways is rendered evident.

Table No. 6, in which are given the properties of saturated steam, taken in connection with Table No. 3, which gives the number of heat units in the water for any given temperature, are of great convenience in all calculations relating to steam-boiler performance.

Table No. 7.— Heat and Work Required to Generate One Pound of Saturated Steam at 212° from Water at 32°.

DISTRIBUTION OF HEAT.	Units of Heat.	Mechanical Equivalent, in Foot-Pounds.
The Sensible Heat.		
1. To raise the temperature of the water from 32° to 212°	180.9	140,740
The Latent Heat.		
2. In the formation of steam	894.0	695,532
3. In expansion against the atmospheric pressure	71.7	55,783
Total heat and work	1,146.6	892,055

Flow of Steam. — When steam is discharged into the atmosphere the velocity of outflow at constant density (when the absolute pressures are greater than 1.73 times the atmospheric pressure) is as given in the accompanying Table No. 8.

Table No. 8.— Outflow of Steam into the Atmosphere.

Absolute Initial Pressure per sq. in. Pounds.	Velocity of Outflow at Constant Density. Feet per Second.	Actual Velocity of Outflow, Expanded Feet per Second.	Discharge per sq. in. of Orifice per min. Pounds.	Horse Power per sq. in. of Orifice if H.P.=30 lbs. per hr. H.P.	Absolute Initial Pressure per sq. in. Pounds.	Velocity of Outflow at Constant Density. Feet per Second.	Actual Velocity of Outflow, Expanded Feet per Second.	Discharge per sq. in. of Orifice per min. Pounds.	Horse Power per sq. in. of Orifice if H.P.=30 lbs. per hr. H.P.
25.37	863	1,401	22.81	45.6	90.	895	1,454	77.94	155.9
30.	867	1,408	26.84	53.7	100.	898	1,459	86.34	172.7
40.	874	1,419	35.18	70.4	115.	902	1,466	98.76	197.5
50.	880	1,429	44.06	88.1	135.	906	1,472	115.61	231.2
60.	885	1,437	52.59	105.2	155.	910	1,478	132.21	264.4
70.	889	1,444	61.07	122.1	165.	912	1,481	140.46	280.9
75.	891	1,447	65.30	130.6	215.	919	1,493	181.58	363.2

The external pressure per square inch has been taken as that existing under the standard atmospheric pressure of 29.921 inches of mercury, — namely, 14.7 pounds absolute, — while the ratio of expansion in the nozzle itself has been taken as 1.624.

When steam flows through pipes its velocity is necessarily decreased by the friction engendered. The resistances presented by bends and valves seriously retard the flow. The following Table No. 9, calculated by an approximate formula, gives the flow of steam through ordinary steam pipes as measured in pounds under a loss of one pound pressure.

In the case of pipes below six inches, the sizes are the commercial and not the actual internal diameters. The calculated flow has in each instance been figured for a pipe having a length 240 times its diameter. The flow varies as the square root of the length of the pipe.

Table No. 9.—Weight of Steam in Pounds per Minute that will Flow through Pipes of Given Diameter with Loss of One Pound of Pressure.

Initial Gauge Pressure, in lbs. per sq. in.	Diameter of Pipe, in Inches. Length of Each = 240 Diameters.										
	¾	1	1½	2	2½	3	4	5	6	8	10
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6

Steam-Pipe Coverings.—The condensation of steam inside an unprotected steam pipe is dependent upon the temperature of the steam within, the temperature of the air without and the velocity of movement of that air. Under all ordinary condition the condensation is so great as to warrant considerable expenditure for its prevention, which may be accomplished to a greater or less extent by applying to the exterior of the pipe proper coverings having the minimum facility for conveying heat. Such coverings depend not only upon the material of which they are constructed,—which should evidently be non-combustible,—but largely upon the air which is held between the particles of the substance. A material which is non-combustible and at the same time of a porous or spongy nature, with numerous air cells or spaces, is naturally adaptable as a covering.

The relative value of such a covering is most readily expressed by the number of heat units it will transmit under given conditions; the lower its conductivity, the higher its efficiency. The results of very carefully conducted tests by Mr. C. L. Norton, for the Massachusetts Manufacturers' Mutual Fire Insurance Company, are presented in Table No. 10, and serve to indicate the relative values of familiar steam-pipe coverings.

Table No. 10. — Steam-Pipe Coverings.

Name of Covering.	Thickness, in Inches.	Weight, in Ounces per Square Foot.	Temperature Corresponding to 10 lbs. Steam Pressure.		Temperature Corresponding to 200 lbs. Steam Pressure.	
			B. T. U. Loss per sq. ft. of Pipe per Minute.	Ratio of Heat Loss to Loss from Bare Pipe.	B. T. U. Loss per sq. ft. of Pipe per Minute.	Ratio of Heat Loss to Loss from Bare Pipe.
Nonpareil Cork.	0.90	21	1.44	0.232	3.04	0.254
Magnesia.	1.12	24	1.59	0.262	3.40	0.284
Air Cell No. 1.	1.12	23	——	——	3.58	0.300
Air Cell No. 2.	1.25	36	1.58	0.261	3.40	0.284
Magnabestos.	1.12	48	2.32	0.383	3.84	0.321
Fire Felt.	1.00	46	2.40	0.395	3.99	0.333
Calcite.	1.25	29	——	——	5.02	0.423
Bare Pipe.	——	——	6.06	1.000	11.96	1.000

CHAPTER III.

COMBUSTION.

Definition. — Although in its broadest sense the term "combustion" applies to all forms of chemical union, in its common acceptance it has reference only to the process of burning, whereby a substance unites with oxygen, with the resulting phenomena of light and heat. A combustible may, therefore, be defined as a substance capable of combining rapidly with oxygen so as to produce light and heat, while oxygen may be classed as a supporter of combustion.

Carbon. — Of all combustibles carbon is the most widely distributed, and readily obtained in nature. Because of its abundance as a constituent of coal, wood, peat, mineral oil and natural gas, these substances are almost exclusively adopted as fuels. To these may be added coke, charcoal and fuel gas, which are produced by special processes from these natural substances. Carbon itself is an infusible, non-volatile solid, of which three distinct modifications occur; viz., (1) diamond, (2) plumbago, or graphite, (3) charcoal, or lampblack. Among natural fuels, — that is, those not prepared by artificial means, — anthracite coal most nearly approaches to the condition of pure carbon, and is to be classed between graphite and charcoal.

Oxygen. — Although oxygen, the universal supporter of combustion, as here defined, is the most abundant of all natural substances, it never exists by itself in nature, but always in association with some other substance. Thus it is that as a constituent of atmospheric air it is associated with the inert gas, nitrogen; the relative proportions of the two gases in pure air remaining practically constant under all conditions.

As determined by recent investigation, pure air is composed, by volume, of —

Oxygen	0.213	parts
Nitrogen	0.787	"
	1.000	

and by weight of —

Oxygen	0.236	parts
Nitrogen	0.764	"
	1.000	

In nature, however, this proportionate composition of pure air is slightly affected by the presence of aqueous vapor, carbonic acid and other impurities. Unless extreme accuracy is desired, it is usually convenient to consider the atmosphere as composed of one volume of oxygen and four volumes of nitrogen. Air is, however, but a mechanical mixture of the two gases, and the oxygen is, therefore, free, without chemical dissociation, to leave the nitrogen and unite with other substances, which it does with great avidity under favorable circumstances. In its independent state, oxygen is colorless, tasteless and slightly heavier than air, in the proportion of 1 to 1.1056.

The Atomic Theory.—A clear understanding of the atomic theory is necessary to a full comprehension of the principles of combustion. This theory, which has been developed through years of investigation, is now universally accepted as the explanation of all chemical phenomena.

By experiment it has been demonstrated that all chemical combinations between elementary substances are made in definite and invariable proportions. For instance, if hydrogen and oxygen be mixed and caused to form water, as already described, it will be found that the entire amount of these two gases will be utilized and enter into combination, only when they originally existed in the exact proportion of two volumes of hydrogen to one volume of oxygen. Observation also proves that if this water is maintained in its gaseous condition it will occupy only the space of two volumes, although for its production a total of three volumes was supplied.

Two volumes of hydrogen and one and a half volumes of oxygen cannot be made to chemically combine to form the compound water, for the hydrogen will unite with only its proportional quantity, leaving the extra half-volume of oxygen unassociated. No matter how large or how small these volumes may be, the same relation holds. It is, therefore, reasonable to suppose that, if the smallest conceivable particle of oxygen be brought into union with two of the smallest conceivable particles of hydrogen, there will be the same result and a minute particle of water will be formed.

These minute particles, the smallest in which elementary substances may be conceived to enter into combination with each other, are called atoms, while the individual particles resulting from their union are known as molecules. From the above reasoning it would appear probable that equal volumes of the elementary gases, at least, contain the same number of atoms, and, therefore, that the atoms are of equal size. Although attempts have been made to calculate the probable dimensions of these atoms, we have no direct knowledge as to their size.

Chemists have adopted as designating symbols for the various elements the initials of their names, followed, when necessary for distinction, by a succeeding letter. Thus hydrogen is designated by H and oxygen by O. The compound water, formed by the chemical union of two atoms of hydrogen and one atom of oxygen, can, therefore, be simply represented by H_2O, the suffix "2" being employed to indicate the presence by volume of twice as much hydrogen as oxygen.

Upon the assumption that the atoms are of equal size, the determination of the relative weights of equal volumes of these two gases, under the same pressure and temperature, is equivalent to determining the relative weights of the atoms themselves; that is, their *atomic weights*. The weight of hydrogen, which is the lightest of all known substances, is taken as unity, the relative weight of oxygen being 16. That is, a given volume of oxygen weighs 16 times as much as an equal volume of hydrogen.

The symbol H_2O, therefore, reveals still another fact as to the composition of water; namely, that 2 atoms of hydrogen, weighing relatively $2 \times 1 = 2$, are combined with 1 atom of oxygen weighing 16. In other words, that by weight, water is composed of 2 parts of hydrogen and 16 parts of oxygen; or more simply, that the ratio of the hydrogen to the oxygen is as 1 to 8.

But in this process of combination it has already been shown that the two volumes of hydrogen and one volume of oxygen unite to form only two volumes of water in its gaseous state, which two volumes represent the space originally occupied by the hydrogen. Hence it is evident that the compound now weighing 18 occupies the same space as an amount of hydrogen weighing 2, and that its relative density is $\frac{18}{2} = 9$. In other words, gaseous water of given temperature and pressure weighs nine times as much as an equal volume of hydrogen under the same conditions.

The common elementary substances entering into the composition of fuel, with their symbols and atomic weights in round numbers, are given in Table No. 11.

Table No. 11.—Symbols and Atomic Weights of Elementary Substances concerned in Combustion.

Name.	Symbol.	Atomic Weight.
Hydrogen.	H.	1
Carbon.	C.	12
Nitrogen.	N.	14
Oxygen.	O.	16
Sulphur.	S.	32

Union of Carbon and Oxygen. — Many elements enter into chemical combination with each other in more than one proportion. This is true of carbon and oxygen. If a piece of carbon, heated to incandescence, be placed in a sufficient volume of oxygen or air, each atom of the carbon will unite with two atoms of the oxygen, to form a compound known as carbonic acid, or carbonic dioxide, the symbol of which is CO_2; the process being indicated by the formula $C + 2O = CO_2$. No matter how plenteous the oxygen, it cannot be made to enter into combination with the carbon in a proportion greater than two atoms of oxygen to one atom of carbon.

This gas is, therefore, evidently a product of complete combustion, there having been a full supply of oxygen. As is shown by Table No. 11, the single atom of carbon weighs 12 relatively to each atom of oxygen which weighs 16; that is, the compound consists by weight of 12 parts of carbon, and $2 \times 16 = 32$ parts of oxygen.

Carbonic acid gas is transparent and colorless, about one and a half times heavier than air, and of a slightly acid taste and smell. It is incombustible, being already the product of complete combustion, and, although not directly poisonous, is neither a supporter of animal life nor of combustion.

If, in turn, this gas, without the accompaniment of sufficient oxygen, be brought into contact with incandescent carbon, it will be deprived of one-half its oxygen, each atom of oxygen thus released uniting with an atom of carbon to form a new compound known as carbonic oxide, with the symbol CO. The process of combination may be symbolically expressed thus: $CO_2 + C = 2CO$, showing that not only is the new compound formed by union of carbon with the released oxygen, but that the carbonic acid thus deprived of its oxygen is thereby also reduced to carbonic oxide. The relative weights of the elements of this compound are, evidently, carbon = 12, and oxygen = 16.

This gas is slightly lighter than air, transparent, colorless and practically odorless, and is destructive to animal life, being, in fact, a direct poison. It is not a supporter of combustion, but, being the product of imperfect combustion, is itself a combustible and may be readily burned in the air. Such being the case, we should expect that the process of burning — which has already been defined as the rapid chemical union of a combustible with oxygen — would result in an accession of oxygen to the carbonic oxide. Experiment will prove this to be true, the product being carbonic acid, the same compound already shown to be the result of complete combustion. Symbolically, the process is expressed by $CO + O = CO_2$. In tabular form, the general properties of carbonic oxide and carbonic acid are shown in Table No. 12, upon the succeeding page.

Table No. 12. — Properties of Carbonic Oxide and Carbonic Acid.

Name.	Symbol.	COMPOSITION.					
		By Weight.			Percentage.		
		Carbon.	Oxygen.	Total.	Carbon.	Oxygen.	Total.
Carbonic oxide,	CO	12	16	28	42.86	57.14	100
Carbonic acid,	CO_2	12	32	44	27.27	72.73	100

Combustion of Fuel. — The two elements contributing most largely to the economic value of any fuel, as measured by its heating power, are carbon and hydrogen. These elements exist in fuels either combined, or, upon the application of heat, associate themselves in a series of complex compounds known as hydro-carbons, the simplest of the list of some fifty being marsh gas, represented by the symbol CH_4. Such portion of the carbon, or hydrogen, as does not thus enter into combination, and for which there exists in the entire substance no further material for combination, is designated as fixed.

Besides these primary elements, fuels usually contain small amounts of oxygen, nitrogen and sulphur, together with a certain percentage of incombustible matter which remains as ash after the process of combustion is complete. The phenomena attendant upon the combustion of ordinary fuels are, therefore, much more complex than those resulting from the combustion of carbon alone. Although it must be evident that fuels of the same general character vary considerably in the proportions of their constituents, their relative average elementary composition by weight is substantially as shown in Table No. 13. The results there given are those determined by ultimate analysis. For the general purposes of comparison of fuels, the method of proximate analysis, whereby only the relative percentages of carbon, volatile matter, ash and moisture are ascertained, is sufficiently refined.

Obviously, owing to the conditions under which combustion takes place, it is impossible to determine in detail the exact order of the process. It is certain, however, that the final results of perfect combustion of ordinary fuel should be carbonic acid gas (CO_2), water (H_2O), nitrogen (N), and possibly a little sulphurous acid (SO_2). The process may be outlined as follows: If, for instance, coal of bituminous character be thrown upon a glowing fire, the heat first volatilizes and frees the hydro-carbons, at comparatively low temperature. These inflammable gases are thereupon immediately ignited, and by the heat thus produced assist in bringing the remainder of the coal to a state of incandescence. The burning of the hydro-carbons is indicative of their union with oxygen, whereby these compounds are broken up and new combinations of a simpler character are formed. The three elements thus presented for combination are carbon, hydrogen and oxygen. If the supply of oxygen is sufficient, the carbon leaves the hydrogen with which it has been associated, and unites with the oxygen to form carbonic acid, an evidence that combustion is complete. The liberated hydrogen also unites with oxygen, if a sufficiency is present, and forms water, which, being at a high temperature, is maintained in its gaseous condition. If, upon dissociation, a portion of the carbon which is liberated in incandescent particles does not immediately meet with its complement of oxygen, it is liable to become cooled to such an extent by the surrounding gases that,

Table No. 13.—Composition of Fuels.

DESCRIPTION.	Carbon.	Hydrogen.	Oxygen.	Nitrogen.	Sulphur.	Ash.
ANTHRACITES.						
France,	90.9	1.47	1.53	1.00	0.80	4.3
Wales,	91.7	3.78	1.30	1.00	0.72	1.5
Rhode Island,	85.0	3.71	2.39	1.00	0.90	7.0
Pennsylvania,	78.6	2.5	1.7	0.8	0.4	14.8
SEMI-BITUMINOUS.						
Maryland,	80.0	5.0	2.7	1.1	1.2	8.3
Wales,	88.3	4.7	0.6	1.4	1.8	3.2
BITUMINOUS.						
Pennsylvania,	75.5	4.93	12.35	1.12	1.10	5.0
Indiana,	69.7	5.10	19.17	1.23	1.30	3.5
Illinois,	61.4	4.87	35.42	1.41	1.20	5.7
Virginia,	57.0	4.96	26.44	1.70	1.50	8.4
Alabama,	53.2	4.81	32.37	1.62	1.30	6.7
Kentucky,	49.1	4.95	41.13	1.70	1.40	7.2
Cape Breton,	67.2	4.26	20.16	1.07	1.21	6.1
Vancouver's Island,	66.9	5.32	8.76	1.02	2.20	15.8
Lancashire gas-coal,	80.1	5.5	8.1	2.1	1.5	2.7
Boghead cannel,	63.1	8.9	7.0	0.2	1.0	19.8
LIGNITES.						
California brown coal,	49.7	3.78	30.19	1.0	1.53	13.8
Australian brown coal,	73.2	4.71	12.35	1.11	0.63	8.0
PETROLEUMS.						
Pennsylvania, crude,	84.9	13.7	1.4	——	——	——
Caucasian, light,	86.3	13.6	0.1	——	——	——
Caucasian, heavy,	86.6	12.3	1.1	——	——	——
Refuse,	87.1	11.7	1.2	——	——	——

when it reaches an abundance of oxygen, its temperature will be too low to permit of chemical union. It will, therefore, pass off as unconsumed and visible carbon, in the form of smoke.

By the time the coal has become incandescent all of the hydro-carbons will have been expelled, and the carbon will be in a condition to enter into combination with the oxygen of the air, or of any surrounding carbonic acid. If oxygen is present in excess, the product will be carbonic acid; but if carbonic acid be brought into contact with the glowing coal, carbonic oxide will be the result. This, in turn, will burn to carbonic acid, if only supplied with sufficient air.

In consideration of the number and great variety of interstices existing between the lumps of coal, and of the various stages of combustion to which different portions of the fuel have attained, it is evident that in their passage through the fire many changes must take place in the composition of the gases. Association and dissociation must follow in rapid succession; at one instant an atom of carbon may be combined with two atoms of oxygen to form carbonic acid, while in the next it may have lost one of its atoms of oxygen and have been reduced to carbonic oxide, which, in turn, may come in contact with a sufficiency of oxygen to again form carbonic acid. In just which combination the carbon and oxygen shall leave the fire and pass to the chimney must depend upon the temperature of the gases and the proportion of oxygen at hand.

Air Required for Combustion.—The definite proportions in which oxygen unites with hydrogen and carbon to form water and carbonic acid—the results of perfect combustion—have already been shown. Expressed in pounds, one pound of hydrogen requires eight pounds of oxygen for its complete combustion; for the atomic weights are respectively $H = 1$ and $O = 16$, and the composition of water, which is the product of perfect combustion of hydrogen and oxygen, is symbolically indicated by H_2O. Hence, substituting atomic weights for symbols, $H_2O = 2 + 16$, and $H_2 : O :: 2 : 16 = H_2 : O :: 1 : 8$.

For the complete combustion of one pound of carbon there are required $2\frac{2}{3}$ pounds of oxygen, for, by atomic weights, $C = 12$ and $O = 16$; hence, $CO_2 = 12 + (2 \times 16)$ and $C : O_2 :: 12 : 32 = C : O_2 :: 1 : 2\frac{2}{3}$.

Air consists, by weight, of 0.236 parts oxygen; therefore, the amount of air required for the combustion of one pound of carbon must be the amount that would contain $2\frac{2}{3}$ pounds of oxygen; that is, $2\frac{2}{3} \div 0.236 = 11.3$ pounds.

In Table No. 14 are presented the principal data regarding oxygen, nitrogen and the elements of the common combustibles, together with the amount of oxygen and air required for each as calculated in the manner just described.

As already shown in Table No. 13, oxygen enters to a certain extent into the original composition of all fuels. In the process of combustion, this oxygen unites with its equivalent of hydrogen, which is thus rendered inert, so far as combination with extraneous oxygen is concerned. In calculation, therefore, this quantity of hydrogen is disregarded, and there are left to be considered only the remaining carbon and hydrogen.

Table No. 14. — Combustion Data.

COMBUSTIBLE.				PRODUCT OF COMBUSTION.				REQUIRED PER POUND OF COMBUSTIBLE.		
NAME.	Symbol.	Atomic or Molecular Weight.	Density or Weight of One Cubic Ft.	NAME.	Symbol.	Atomic or Molecular Weight.	Density or Weight of One Cubic Ft.	Oxygen.	Air.	
			Pounds.				Pounds.	Pounds.	Pounds.	Cubic Feet at 62°.
				Oxygen,	O	16	0.08928			
				Nitrogen,	N	14	0.07837			
Hydrogen,	H	1	0.00559	Water,	H_2O	18	——	8.00	33.9	444
Carbon,	C	12	——	Carbonic ox.	CO	28	0.07806	1.33	5.7	75
Carbon,	C	12	——	Carbonic acid	CO_2	44	0.12341	2.67	11.3	148
Carbonic ox.	CO	28	0.07806	Carbonic acid	CO_2	44	0.12341	0.57	2.41	32
Marsh gas,	CH_4	16	0.04464	Water,	H_2O	18	——	4.00	16.9	222
				Carbonic acid,	CO_2	44	0.12341			
Olefiant gas,	C_2H_4	28	0.07809	Water,	H_2O	18	——	3.43	14.5	190
				Carbonic acid,	CO_2	44	0.12341			
Sulphur,	S	32	——	Sulphurous acid,	SO_2	64	0.17860	1.00	4.25	56

The method of calculation of the amount of air necessary for the combustion of ordinary coal can best be explained by means of an example based upon the known composition of a certain fuel, as, for instance, that of the Maryland semi-bituminous coal, given in Table No. 13. For simplicity these proportionate figures are here given in pounds instead of in per cent.

Carbon	80.0 pounds.
Hydrogen	5.0 "
Oxygen	2.7 "
Nitrogen	1.1 "
Sulphur	1.2 "
Ash	8.3 "

The nitrogen is inert, the sulphur, because of the small amount in which it is present, may be disregarded, and the ash, being incombustible, has no effect on the result.

We may, therefore, estimate as follows for the amount of oxygen required: The 2.7 pounds of oxygen will render inert $\frac{2.7}{8} = 0.3375$ pounds of hydrogen, for it will directly combine with that amount. The constituents to be considered thus become —

Carbon	80	pounds.
Hydrogen 5.0 — 0.3375 = ...	4.6625	"
	84.6625	pounds.

and their requirements in the way of oxygen will be —

Carbon 80 × 2⅔ = ...	213.33	pounds.
Hydrogen 4.6625 × 8 = ...	37.30	"
	250.63	pounds.

The weight of air containing the above quantity of oxygen is $\frac{250.63}{0.236} = 1061.57$ pounds. The quantity required per pound of combustible will, therefore, be $\frac{1061.57}{84.6625} = 12.54$ pounds. As these constituents are part of a quantity of coal weighing 100 pounds, when its elementary moisture is included, the amount of air necessary, per pound of coal, is $\frac{1061.57}{100} = 10.62$ pounds.

For approximate calculation of the weight of air required for combustion the following formula may be used: —

$$\text{Weight of air} = 12C + 36\left(H - \frac{O}{8}\right).$$

In this equation, the weights of carbon, hydrogen and oxygen are represented respectively by their symbols, C, H and O, the amount of hydrogen rendered inert by the oxygen in the fuel is allowed for, and the proportion of oxygen and nitrogen in the atmosphere is taken as one to four.

The air required per pound of fuel, for various fuels of typical composition, as calculated by the above formula, is shown in Table No. 15; the weights of the elements and of the air being given in pounds.

As is evident by what follows, it is unnecessary, for practical purposes, to compute with great exactness the weight of air necessary for the combustion of fuel; for the excess of air which is usually supplied, together with the variableness in the composition of fuel, renders all ordinary calculations somewhat approximate. It is, therefore, the common practice to estimate the approximate amount required, for either coke or coal, at 12 pounds per pound of fuel.

Table No. 15.—Air Required for Combustion of Fuels.

Fuel.		Weight of Given Constituent in One Pound of Fuel.			Air Required per Pound of Fuel. Pounds.
		Carbon.	Hydrogen.	Oxygen.	
CHARCOAL—	From wood,	0.93	——	——	11.6
"	From peat,	0.80	——	——	9.6
COKE—	Good,	0.94	——	——	11.28
COAL—	Anthracite,	0.915	0.035	0.026	12.13
"	Dry bituminous,	0.87	0.05	0.04	12.06
"	Coking,	0.85	0.05	0.06	11.73
"	Coking,	0.75	0.05	0.05	10.58
"	Cannel,	0.84	0.06	0.08	11.88
"	Dry, long-flaming,	0.77	0.05	0.15	10.32
"	Lignite,	0.70	0.05	0.20	9.30
PEAT—	Dry,	0.58	0.06	0.31	7.68
WOOD—	Dry,	0.50	——	——	6.00
MINERAL OIL,		0.85	——	——	15.65

Air for Dilution.—The preceding calculations of air supply are based upon the assumption that each individual atom of oxygen in the air comes in contact and unites with its proportion of hydrogen or carbon in the fuel. When it is considered that this oxygen is intimately united with about four times its volume of nitrogen, whereby it is to a certain extent separated from the fuel, and, further, that the variety in the arrangement of the fuel and the passages through it affects any attempt at equal distribution of the air, it must be evident that the above assumption cannot ordinarily be maintained in practice. It, therefore, usually becomes necessary in practice to furnish sufficient air in excess of the calculated amount to insure complete combustion in all parts of the furnace.

Evidently, the amount of air supplied for dilution must vary greatly in different cases. This is clearly shown by the results of numerous careful tests of different boilers by Messrs. Donkin and Kennedy. In each case the volume of air supplied was determined by chemical analysis of the furnace gases, the results of which, together with the deductions relating to the amount of air supplied, are presented in Table No. 16. It will be noted that the dry air supplied, per pound of coal, ranges from 16.1 pounds to 40.7 pounds, and that the corresponding ratio of air used to air theoretically required ranges from 1.56 to 4.28; that is, from 56 per cent to 328 per cent in excess. The composition of the gases is given by weight.

Table No. 16. — Analysis and Calculations Relating to Furnace Gases and Air Supply.

ANALYSIS AND CALCULATIONS.	NUMBER OF TEST.							
	I	II	III	IV	V	VI	VII	VIII
Per cent of CO_2,	15.15	13.00	18.21	11.71	7.90	10.44	11.00	14.95
Per cent of CO,	2.59	0.00	0.24	0.00	0.00	0.28	0.10	0.28
Per cent of O,	6.46	11.15	7.55	13.13	17.00	13.37	13.20	8.31
Per cent of N,	75.80	75.85	74.00	75.16	75.10	75.90	75.70	76.46
Pounds dry air per pound of C,	18.5	27.6	19.2	30.7	46.0	33.1	32.3	23.3
Pounds dry air per pound of coal,	16.4	24.4	17.0	27.2	40.7	29.3	28.6	20.6
Ditto, per pound pure dry coal,	16.9	25.2	17.5	28.0	42.2	30.3	29.6	21.2
Pounds dry furnace gases per pound pure dry coal,	17.5	25.8	18.1	28.6	42.8	30.9	30.2	21.8
Ratio of air used to air theoretically required,	1.58	2.40	1.63	2.61	4.28	2.82	2.76	1.98

ANALYSIS AND CALCULATIONS.	IX	XI	XII	XIII	XIV	XVII	XIX	XX
Per cent of CO_2,	8.60	16.50	15.10	17.94	14.90	11.10	11.53	18.88
Per cent of CO,	0.00	0.21	0.00	1.02	0.00	0.00	0.00	0.34
Per cent of O,	14.40	7.76	8.80	6.53	6.60	13.10	13.03	5.85
Per cent of N,	77.00	75.53	76.10	74.51	78.50	75.80	75.44	74.93
Pounds dry air per pound of C,	42.1	21.2	23.7	18.2	24.0	32.7	31.2	18.3
Pounds dry air per pound of coal,	37.3	18.8	21.0	16.1	21.3	29.0	27.6	16.2
Ditto, per pound pure dry coal,	38.6	19.4	21.7	16.7	22.0	30.0	28.6	16.8
Pounds dry furnace gases per pound pure dry coal,	39.2	20.0	22.3	17.3	22.6	30.6	29.2	17.4
Ratio of air used to air theoretically required,	3.6	1.81	2.02	1.56	2.05	2.80	2.67	1.56

Accepting 12 pounds of air, per pound of fuel, as necessary for combustion, the amount required where 100 per cent is supplied for dilution, as in the case of natural draft and hand-firing, will be 24 pounds. But with forced draft the quantity of air required for dilution, as stated by Rankine, "is certainly much less than that which is required in furnaces with chimney draft; and there is reason to believe that on an average it may be estimated at about *one-half* of the air required for combustion." That is, the total amount would be 18 pounds.

This applies where hand-firing is the practice. But when, through the action of a properly applied mechanical stoker supplied with air under pressure, as by means of a fan, the bed of fuel is constantly maintained in the most suitable condition for utilizing the air supplied, the amount required for dilution is reduced to a minimum. This is particularly true when the stoker grate provides special advantages for the equable distribution of the air. Recent tests, by Mr. J. M. Whitham,[1] show, not only the decreased air supply necessary with a good mechanical stoker, but also the reduction in the amount of air required per pound of fuel when a high rate of combustion is maintained by the use of forced draft. With a combustion of twelve pounds of buckwheat coal per square foot of grate per hour, the air was found to be 85.6 per cent in excess of that chemically required; while with a rate of 45.4 pounds almost perfect evaporative efficiency was secured when there was an actual deficiency of 11.2 per cent in the air supply below the chemical requirements. Startling as this result appears, it is reported by an able expert engineer. It certainly points toward the possibilities of reduced air supply with mechanical draft.

"In almost all large boiler furnaces," as stated by Prof. H. B. Gale,[2] "a material improvement in economy may be made by cutting down the grate surface and employing forced draft. Theoretically, 12 pounds of air are sufficient to completely burn a pound of average coal; but in practice, with large grate surfaces and weak draft, between 20 and 30 pounds are required. By the employment of forced draft and judicious proportioning of the furnace, the quantity of air may be reduced easily to 18 pounds, with the result of a white heat in the furnace and better combustion, besides the saving of a great part of the expense of a high chimney."

As the weight of dry air at 62° is 0.0761 pounds per cubic foot, the volumes corresponding to the above weights are as indicated in Table No. 17.

Table No. 17. — Amount of Air Required for Combustion.

	Without Dilution.	With 50 per cent Dilution.	With 100 per cent Dilution.
Weight of air,	12 pounds.	18 pounds.	24 pounds.
Volume of air, exact,	157.7 cu. ft.	236.5 cu. ft.	315.4 cu. ft.
Volume of air, in round numbers,	150 cu. ft.	225 cu. ft.	300 cu. ft.

The latter figures, given in round numbers, are those usually employed.

An insufficient supply of air causes imperfect combustion of the fuel, which in bituminous coal is indicated by the production of smoke, and in coke and anthracite coal by the discharge of carbonic oxide from the chimney. An excess of air causes waste of heat to the amount corresponding to the weight of air in excess of that which is necessary, and to the elevation of temperature at which it is discharged from the chimney above that of the external air. Obviously, the maximum efficiency to be secured in the process of combustion is to be sought between these two extremes.

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[1] Experiments with Automatic Mechanical Stokers. J. M. Whitham, Trans. Am. Soc. of Mech. Engineers, Vol. XVII.

[2] Coal as a Source of Power. H. B. Gale. A paper read before the California Electrical Society, May 15, 1893.

Analysis of Flue Gases. — It is a comparatively simple matter, by means of the proper apparatus, to determine from samples the relative proportions of carbonic oxide, carbonic acid and oxygen in the gases leaving a boiler furnace. An apparatus of this character, devised by Orsat, consists of three pipettes in connection with a graduated burette for measuring the volumes of gas, and a pressure bottle to control the movement of the gases undergoing analysis. The pipettes contain respectively potassium hydrate for the absorption of carbonic acid, an alkaline solution of potassium pyrogallate to absorb the oxygen, and cuprous chloride to absorb the carbonic oxide. By means of the pressure bottle the sample of flue gas is forced through the pipettes in the order named, and the amount absorbed by each is measured by means of the burette. The amounts, by volume, thus obtained may be readily transformed into amounts by weight by multiplying by the densities of the various gases. Although such an analysis does not directly determine the amount of nitrogen present in the flue gases, yet its actual amount, as well as that of the air supply, may be readily ascertained by calculation.

To illustrate the method of calculation, take, for instance, the result of an analysis showing 11.5 per cent of carbonic acid, 0.9 per cent of carbonic oxide, and 7.4 per cent of free oxygen, all by volume. Evidently, the nitrogen being the only other constituent of the flue gases which is of importance, it must be present in sufficient quantity to make up the unit volume of gas. Its volume will, therefore, be —

$$100 - (11.5 + 0.9 + 7.4) = 80.2 \text{ per cent.}$$

In the calculation of the weight of nitrogen and of air supply, it is convenient to treat the percentages by volume as the number of cubic feet of the several gases in 100 cubic feet of flue gas. Referring to Table No. 14 for the proper volumes, as therein given, the composition of the flue gas by weight appears to be —

Gas.	Volume.	Density.	Weight.
Carbonic acid,	11.5	0.12341	1.4192
Carbonic oxide,	0.9	0.07806	0.0703
Oxygen,	7.4	0.08928	0.6607
Nitrogen,	80.2	0.07837	6.2853

As the atomic weights of carbon and oxygen are respectively 12 and 16, it is evident, as is shown by the following simple calculation, that in one pound of carbonic acid the carbon constitutes —

$$\frac{2\times16}{12+(2\times16)}=\frac{32}{44}=\frac{8}{11}$$

of the weight, the remaining $\frac{3}{11}$ being oxygen. In a similar manner, it appears that one pound of carbonic oxide is composed of —

$$\frac{16}{12+16}=\frac{16}{28}=\frac{4}{7}$$

of a pound of oxygen and $\frac{3}{7}$ of a pound of carbon. Therefore, the weight of oxygen in a pound of the above-stated flue gases would be —

In the carbonic acid, $\frac{8}{11}\times1.4192=$.	1.0322 pounds.
In the carbonic oxide, $\frac{4}{7}\times0.0703=$.	0.0402 "
Free oxygen	0.6607 "
Total weight of oxygen . .	1.7331 pounds.

and the weight of carbon would be —

In the carbonic acid, $\frac{3}{11}\times1.4192=$.	0.3870 pounds.
In the carbonic oxide, $\frac{3}{7}\times0.0703=$.	0.0301 "
Total weight of carbon . . .	0.4171 pounds.

As the air consists, by weight, of 0.236 parts of oxygen, the above-estimated weight of oxygen would be contained in —

$$\frac{1.7331}{0.236}=7.36 \text{ pounds of air,}$$

and the supply of air per pound of carbon, the combustion of which resulted in flue gases having the composition given upon the preceding page, must, therefore, have been —

$$\frac{7.36}{0.4171}=17.65 \text{ pounds.}$$

If the coal from the combustion of which these gases resulted had contained 85 per cent of carbon, 3.7 per cent of hydrogen and 2.4 per cent of oxygen, the air supply per pound of coal would be calculated as follows: The supply of air per pound of coal, disregarding the oxygen and hydrogen present therein, would be —

$$0.85\times17.65=15.00 \text{ pounds.}$$

But, on the basis already established, that the oxygen in the fuel renders inert one-eighth of its weight of hydrogen, and the remnant is available for combustion, there would be added to the air per pound of coal —

$$36\left(0.037 - \frac{0.024}{8}\right) = 0.468 \text{ pounds,}$$

making the total air supply per pound of coal, —

$$15.00 + 0.468 = 15.468 \text{ pounds.}$$

Heat of Combustion.— As determined by the most recent and refined calorimetric tests, the heat of combustion, as measured by the number of British thermal units that are given out upon the combustion of one pound of a given substance, is for each of the following —

Carbon burned to CO_2	14,650	B. T. U.
Carbon burned to CO	4,400	"
Hydrogen	62,100	"
Marsh gas	23,513	"
Olefiant gas	21,343	"
Carbonic oxide burned to CO_2 . . .	4,393	"

The great loss of heat, due to the incomplete combustion of carbon, is clearly presented in the differences between the total heat of perfect combustion of carbon to CO_2 (viz., 14,650 B. T. U.), and that of carbon to CO (viz., 4,400 B. T. U.); the latter being the product of incomplete combustion as already stated in a previous section.

One pound of carbon, when imperfectly burned, produces $\frac{12 + 16}{12} = 2\frac{1}{3}$ pounds of carbonic oxide. If this quantity of gas be burned to form carbonic acid, the total amount of heat given out will be $14{,}650 - 4{,}400 = 10{,}250$ B. T. U.; showing that ultimately the carbon gives out its full heat value, no matter what the order of formation of the carbonic acid may have been, whether by direct union of carbon and oxygen, or through the intermediate agency of the carbonic oxide. As the 10,250 B. T. U. are given out by 2⅓ pounds of carbonic oxide, its heat value per pound is, evidently, $\frac{10{,}250}{2\frac{1}{3}} = 4{,}393$ B. T. U.

In calculating the heat of combustion of a fuel, it is customary to disregard, as already explained, that portion of the hydrogen for which there exists in the fuel a sufficient amount of oxygen to form water. The remainder of the carbon and hydrogen thus becomes available for producing heat, and may be introduced in an approximate formula, based upon that for estimating the air required for combustion. Disregarding the effect of inherent nitrogen and sulphur, this formula may be thus expressed: —

$$\text{Heat, in B. T. U.,} = 14{,}650\ C - 62{,}100\ (H - \frac{O}{8}),$$

in which the weights of carbon, hydrogen and oxygen in one pound of fuel are respectively represented by their symbols, C, H and O.

This formula, applied to the determination of the heat of combustion of the Maryland semi-bituminous coal in Table No. 13, appears as follows: —

$$\text{Heat, in B. T. U.,} = 14{,}650 \times 0.80 + 62{,}100\ (0.05 - \frac{0.027}{8}) = 14{,}615.$$

Theoretically, the total calorific value, as determined by the calorimeter and as calculated from analysis, should agree. But, on the one hand, there is opportunity for error or imperfection on the part of the calorimeter; while on the other, the formula employed for calculation from the analysis may fail to make due allowance for heat lost in dissociation, or may not properly recognize the influence of minor constituents. In both cases there is great difficulty in obtaining similar samples. This accounts for differences which frequently exist in reported results. Thus, the calorimetric tests of Scheurer-Kestner were about ten per cent, on an average, higher than the analyses; while results reported by Mr. F. W. Dean[1] show the calorific value, as determined by calorimeter, to be about six per cent less than that calculated from the test. A comparison of the results obtained by these two methods of determination is presented in Table No. 18, from the tests of Mahler on various American and foreign coals.

Table No. 18. — Heat of Combustion of Fuels.

Kind of Fuel.	Analysis.					Per cent of Volatile Matter Exclusive of Water and Ash.	Calorific Value Observed. B. T. U.	Calorific Value Excluding Water and Ash. B. T. U.	Calorific Value Calculated. B. T. U.
	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Hydroscopic Water.	Ash.				
Anthracite, from Penna.,	86.456	1.995	2.199	3.450	5.900	3.00	13,471	14,861	15,210
Semi-anth., from Commentry,	84.928	2.892	5.005	1.775	5.400	3.19	14,130	15,221	15,048
Semi-bituminous, from Aniche	85.937	4.198	5.240	0.625	4.000	11.93	15,167	15,901	15,638
Bituminous, from Anzin,	83.754	4.385	5.761	1.100	5.000	21.51	14,492	15,433	15,640
Wigan cannel coal,	78.382	5.060	5.058	0.600	10.900	31.64	13,970	15,682	16,220
Lignite, from Styria,	65.455	4.782	24.303	0.710	4.750	50.34	13,111	11,963	11,898
Coke, Penna. anthracite,	91.036	0.685	2.146	0.233	5.900	68.93	13,550	14,465	14,540

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[1] Transactions Am. Soc. Mech. Engineers, Vol. XVII., p. 285.

In boiler practice, owing to the opportunities for loss of heat through radiation, heat carried off by flue gases, incomplete combustion, etc., the maximum efficiency attainable with the best possible boiler and warm-blast or feed-water heating apparatus appears to be about 90 per cent. Under ordinary conditions with good coal, the efficiency may be assumed to average from 60 to 70 per cent, and with poor coals from 50 to 60 per cent. It is, therefore, customary, for rough figuring, to consider the available heat of combustion per pound of fuel to be ordinarily from 10,000 to 12,000 B. T. U.

The total heat of various fuels will be shown in succeeding tables.

Ideal Temperature of Combustion. — From the known total and specific heats of combustibles may be calculated the temperature which would result from their combustion if all possible losses were prevented. In ordinary practice those losses must occur and the efficiency of fuels be reduced thereby. It is, therefore, impossible to attain in practice the full ideal temperature. The general properties of the substances entering into a discussion of the combustion of fuels are given in Table No. 19.

Table No. 19. — Properties of Substances Concerned in Combustion.

Substance.	Symbol.	Atomic or Molecular Weight.	Specific Volume.	Specific Heat in a Gaseous Condition.	Density or Weight per Cubic Foot. Pounds.
Hydrogen,	H	1	178.881	3.409	0.00559
Carbon,	C	12	——	——	——
Nitrogen,	N	14	12.7561	0.2438	0.07837
Oxygen,	O	16	11.2070	0.2175	0.08928
Carbonic oxide,	CO	12 + 16	12.81	0.2450	0.07806
Carbonic acid,	CO_2	12 + 2 × 16	8.10324	0.2169	0.12341
Water,	H_2O	2 + 16	——	0.4805	——
Air,			12.3909	0.2375	0.08071
Ash,				0.2	

It has already been shown that one pound of carbon, burned to carbonic acid, requires 2⅔ pounds of oxygen. Hence the total product of combustion of one pound of carbon = 3⅔ pounds, as is also evident by the following calculation based upon the atomic weights: —

$$\frac{12 + (2 \times 16)}{12} = 3\tfrac{2}{3} \text{ pounds.}$$

It has further been shown that a total of 11.3 pounds of air is required to furnish 2⅔ pounds of oxygen. Therefore, the total weight of the products or results of combustion must be 12.3 pounds, and the weight of the nitrogen alone 12.3 — 3⅔ = 8.63 pounds. As the specific heat of a substance is a measure of the number of thermal units necessary to raise its temperature through one degree, the total number of units required to raise through one degree the products of combustion of one pound of carbon, with the associated nitrogen, may be determined thus:—

	Weight.		Specific Heat.		B. T. U.
Carbonic acid	3⅔	×	0.2169	=	0.7953
Nitrogen	8.63	×	0.2438	=	2.1040
B. T. U. per degree					2.8993

As one pound of carbon in the process of burning gives out 14,650 B. T. U., and as it requires 2.8993 B. T. U. to raise through one degree the products of combustion, including the accompanying nitrogen, the ideal temperature resulting from the combustion of one pound of carbon must be 14,650 ÷ 2.8993 = 5,053°. In the same manner the ideal temperature of combustion of hydrogen may be calculated, and as it makes no difference in the temperature whether the oxygen required for this union is derived from the original constituents of the fuel or from the atmosphere, the entire amount of hydrogen in the fuel is taken into account.

For the purpose of illustrating the process of calculation, the Maryland semi-bituminous coal in Table No. 13 may be again considered. The important constituents of this coal, expressed in per cent of one pound of coal, are—

Carbon	80.0	per cent.
Hydrogen	5.0	"
Oxygen	2.7	"
Nitrogen	1.1	"

For simplicity the ash and sulphur may be disregarded, and also the latent heat of the steam formed by the combination of hydrogen and oxygen. The heat of combustion of this coal has already been calculated as 14,615 B. T. U. By the process explained in the section on "Air Required for Combustion," it has also been shown that, for the total combustion of the carbon and hydrogen contained in one pound of this coal, there are required 10.62 pounds of air, of which 2.51 pounds will be oxygen and 8.11 pounds will be nitrogen. This amount of nitrogen, added to that already in the coal, makes the total 8.11 + 1.1 = 9.21 pounds.

The total amount of carbonic acid produced by the union of oxygen with 0.8 pounds of carbon is 3⅔ × 0.8 = 2.933 pounds, and as the total products of combustion of one pound of hydrogen are $\frac{2+16}{2} = 9$ pounds, the weight of the products of combustion of the hydrogen in the coal will be $9 \times 0.05 = 0.45$ pounds. Hence, the thermal units required to raise each of these combustibles through one degree are —

	Weight.		Specific Heat.		B. T. U.
Carbonic acid	2.933	×	0.2169	=	0.6362
Water	0.45	×	0.4805	=	0.2162
Nitrogen	9.21	×	0.2438	=	2.2454
Total B. T. U.					3.0978

The ideal temperature of combustion, therefore, appears to be —

$$14{,}615 \div 3.0978 = 4{,}718^\circ.$$

If, for the purposes of dilution, there had been provided 50 per cent of air in excess of that theoretically required for complete combustion, the amount of heat necessary to raise the temperature of the products of combustion through one degree would have been increased, and the final temperature reduced, as is evident from the following: —

	Weight.	Specific Heat.	B. T. U.
50 per cent air for dilution,	$\frac{10.62}{2} = 5.31$	× 0.2375	= 1.2611
Products without dilution as above . . .			= 3.0978
Total B. T. U.			4.3589

and $14{,}615 \div 4.3589 = 3{,}353^\circ$. The cooling effect of the air, which is absolutely necessary for dilution, is thus made evident by a decrease of $4{,}718 - 3{,}353 = 1{,}365^\circ$ when it is only 50 per cent in excess.

While the temperature of combustion of a complex fuel may be calculated with much greater refinement by taking into account all of the minor constituents, the results thus obtained are practically of but little more value than those derived from this approximate method; for local conditions in boiler practice always have considerable effect in reducing the actual temperature to somewhat below the ideal. Mr. J. C. Hoadley,[1] in carefully conducted tests with a water-platinum calorimeter, found in the heart of the fire under an ordinary boiler a temperature of 2,426°, the coal consisting of 82 per cent of carbon, and the supply of air being 21.4 pounds per pound of coal. Immediately above the fire, and at the bridge wall, the temperature rapidly decreased through losses by radiation and conduction to the walls and the water in the boiler, so that the corresponding temperature at the bridge wall was only 1,341°.

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

The ideal temperature of combustion of the Maryland semi-bituminous coal, with different degrees of dilution, as determined by calculation in the manner already indicated, is presented in Table No. 20.

Table No. 20.—Ideal Temperatures of Combustion with Different Degrees of Dilution.

Percentage of Dilution.	Ideal Temperature.	Loss of Temperature due to Dilution.
0	4,718°	
50	3,353	1,365°
100	2,600	2,118
150	2,124	2,594

These figures indicate only the increments of temperature under the given conditions; hence, to obtain the actual thermometric temperature, they must be increased by the initial temperature of the air. Thus, if the air is supplied at 62°, the ideal temperature, with 100 per cent dilution, would be 2,600 + 62 = 2,662°, while if the air had been previously heated by special means to 300°, it would be 2,600 + 300 = 2,900°.

CHAPTER IV.

FUELS.

Definition. — Fuels may be defined as those substances which, by means of atmospheric air, can be economically burned to generate heat. The principal constituent of all is carbon, with which hydrogen is usually associated. They may be broadly classified as natural and artificial.

Natural Fuels. — Natural fuels are such forms of carbon and its compounds with hydrogen as occur distributed in nature, either as products of existing organic life, or as the fossilized remains of a prehistoric growth. Under this heading are included the varieties of wood, coal, mineral oil and natural gas. The solid fuels may be classified as follows: —

- Wood.
- Peat.
- Coal,
  - Lignite.
  - Bituminous,
    - Non-caking, rich in oxygen.
    - Caking.
    - Non-caking, rich in carbon.
  - Anthracite.

Artificial Fuels. — Artificial fuels comprise those forms of carbon or its compounds with hydrogen which owe their origin to some process of manufacture, but are not commonly found distributed in nature. These are generally obtained from natural fuels by some special process; as, for instance, charcoal from wood, coke and volatile hydro-carbons from coal. Artificial fuels include the various attempts to cement together, in the form of blocks or briquettes, such combustible refuse as is too small to be otherwise profitably consumed.

The products of carbonization may be classified as follows: —

- Products of carbonization,
  - Solid,
    - Wood — Charcoal.
    - Peat — Charcoal.
    - Coke.
  - Volatile,
    - Carbonic oxide.
    - Hydrogen.
    - Hydro-carbons.

Wood. — Although the term "wood" broadly includes all substances of vegetable fibre which have not undergone geological changes, it applies directly to the fairly compact substance which constitutes tree trunks and branches. With reference to its heating power, wood under this definition may be classed as hard and soft. Hard woods include the oak, hickory, maple, beech and walnut; and soft woods, the pine, elm, birch, chestnut, poplar and willow. When freshly cut, wood contains nearly fifty per cent of moisture, which seriously reduces its calorific value. Through the process of air or kiln drying, the amount of moisture may be brought down to from 10 to 20 per cent.

The approximate weight of one cord of thoroughly air-dried wood, and its calorific value relatively to soft coal, are given in Table No. 21, for the kinds specified.

Table No. 21. — Weight and Calorific Value of Wood.

KIND OF WOOD.	Weight.	Weight of Coal of Equivalent Calorific Value.
Hickory or hard maple,	4,500 pounds.	1,800 pounds.
White oak,	3,850 "	1,540 "
Beech, red and black oak,	3,250 "	1,300 "
Poplar, chestnut and elm,	2,350 "	940 "
Average pine,	2,000 "	800 "

The above indicates that about 2¼ pounds of dry wood are equal to a pound of average soft coal, and that the calorific value of the same weight of various woods is substantially the same. The average chemical composition of the ordinary kinds of wood, when perfectly dry, is shown by Table No. 22 to be substantially the same.

Table No. 22. — Composition of Wood.

KIND OF WOOD.	COMPOSITION.				
	Carbon.	Hydrogen.	Oxygen.	Nitrogen.	Ash.
Beech,	49.36	6.01	42.69	0.91	1.06
Oak,	49.64	5.92	41.16	1.29	1.97
Birch,	50.20	6.20	41.62	1.15	0.81
Poplar,	49.37	6.21	41.60	0.96	1.86
Willow,	49.96	5.96	39.56	0.96	3.37
Averages,	49.70	6.06	41.30	1.05	1.80

Calculated by the usual formula as already presented, the heat value of wood of the average composition, shown in Table No. 22, would be 7,838 B. T. U. If this wood is considered to be in the ordinary condition of air-dried stock, containing about 20 per cent of moisture, then its heating value per pound would be only three-quarters as much, or 5,879 B. T. U.

Straw and Tan. — Evidently straw can be economically employed as a fuel only where the supply is directly at hand and the cost of other fuel is excessive. Analyses of air-dried straw have shown it to be of the composition indicated in Table No. 23.

Table No. 23. — Composition of Straw.

CONSTITUENTS.	Wheat Straw.	Barley Straw.	Mean.
Carbon,	35.86	36.27	36.0
Hydrogen,	5.01	5.07	5.0
Oxygen,	37.68	38.26	38.0
Nitrogen,	0.45	0.40	0.50
Ash,	5.00	4.50	4.75
Water,	16.00	15.50	15.75

Straw of the mean composition given above has a calorific value, deducting the heat lost in evaporating its constituent water, of 5,155 B. T. U. It weighs, when pressed, 6 to 8 pounds per cubic foot. Oak bark, after having served its purpose as a tanning agent, thereby becoming spent tan and consisting only of the fibrous portion of the bark, is used as a fuel, but only under the economical conditions which hold in the use of straw as a fuel. That is when the tan is readily accessible and its total cost when placed in the furnace is less, for a given result, than that of other available fuels. In the process of tanning the bark loses about 20 per cent of its weight. Perfectly dry tan, containing 15 per cent of ash, has a heating power of 6,100 B. T. U.; while tan containing 30 per cent of water — its usual condition of dryness — has a calorific value of only 4,284 B. T. U. The weight of water evaporated at 212° by one pound of tan under these two conditions of dryness is as follows: —

	Perfectly Dry.	With 30 Per Cent of Moisture.
Water supplied at 62° .	5.46 pounds.	3.84 pounds.
Water supplied at 212° .	6.31 pounds.	4.44 pounds.

The conditions of success in burning tan, as is the case with all wet fuel, consist in completely surrounding it with heated surfaces and burning fuel so that it may be rapidly dried, and then so arranging the apparatus that thorough combustion may be secured.

Bagasse. — The term "bagasse," or megass, is generally understood to apply to that portion of the sugar cane that is left after extracting the juice. As the methods of extraction give results varying all the way from 40 per cent to 80 per cent, it is evident that it includes substances differing greatly in composition. In its broadest sense it may, therefore, be taken as meaning the refuse discharged from the cane mill or diffusion process, whether it comes from a mill giving 40 per cent extraction and leaving 70 per cent of moisture, or whether it be the air-dried bagasse of the tropics with only 10 per cent of moisture.

Mill bagasse is the refuse left after the juice has been extracted by means of the mill rolls. Diffusion bagasse is the material remaining after a series of soaking processes for which it has been chopped into small pieces, and whereby the saccharine matter has diffused itself throughout the mass of water in which the cane has been placed. The original cane, and likewise the bagasse, consist of woody fibre, water and combustible salts; but, naturally, the squeezing process reduces the percentage of liquid matter, and proportionately increases the relative amount of fibrous material in the bagasse. Upon the fibre, which is principally carbon, the value of bagasse as a fuel largely depends. In tropical canes it constitutes in round numbers about 12 per cent of the original cane, while in Louisiana cane 10 per cent is a fair average. Tropical cane and the bagasse therefrom have the composition given in Table No. 24.

Table No. 24. — Composition of Tropical Cane and Bagasse.

CONSTITUENTS.	CANE.	BAGASSE.		
		66 per cent Extraction.	70 per cent Extraction.	72 per cent Extraction.
Woody fibre,	12.5	37	40	45
Water,	73.4	53	50	46
Combustible salts,	14.1	10	10	9

The proportional composition of Louisiana bagasse is clearly shown, for different degrees of extraction, in Table No. 25.

Table No. 25. — Composition of Dry Louisiana Bagasse.

CONSTITUENTS.	Percentage.
Volatile matter,	81.37
Fixed carbon,	14.26
Ash,	4.6

Carefully conducted calorimetric tests of bagasse, when under different conditions, by Dr. W. O. Atwater, give the heat values which are indicated in Table No. 26.

Table No. 26.—Calorimetric Tests of Bagasse.

Description.	Per cent Moisture in Sample.	B. T. U. per Pound as received.	B. T. U. per lb. Dry Matter from Preceding Column.	B. T. U. per lb. Dry Matter by Actual Test.
Purple cane exhaust chips, direct from diffusion battery,	90.36	799	8,288	8,320
Striped cane exhaust chips, direct from diffusion battery,	90.54	873	9,229	8,289
Purple cane exhaust chips, passed through laboratory mill,	73.34	1,966	7,373	8,309
Striped cane exhaust chips, passed through laboratory mill,	69.62	2,547	8,384	8,384
Averages,			8,319	8,325

This table serves to show the serious effect of contained water upon the heat value of the bagasse of different extractions. This effect is to be expected, for all of the water in the bagasse when employed as fuel must be vaporized before the combustible matter can be consumed, and in the process of vaporization an enormous amount of heat is rendered latent, and thus lost to the furnace so far as heating effect is concerned.

The only method available in estimating the fuel value of the different extractions of mill bagasse is that based upon the assumption that bagasse consists of two substances,—fibre and juice,—and that this juice has the same composition as that which has already been extracted. To obtain a heat value for juice, it must be divided into sugar and molasses. Thus, for instance, an average cane consisting of—

Fibre		10	per cent,
Juice	Sucrose	12	"
	Glucose	2	"
	Solids—not sugar,	1	"
	Water	75	"
		100	per cent,

will, upon passing through a mill giving an extraction of 75 per cent, be reduced to bagasse, the weight of which will be only 25 per cent of the original cane of which it formed a part.

The proportional composition in per cent of the weight of the original cane and of the resulting bagasse will then be as presented in Table No. 27.

Table No. 27. — Composition of Mill Bagasse.

CONSTITUENTS.	In per cent of Original Cane.	In per cent of the Resulting Bagasse.
Water,	12.75	51
Fibre,	10.0	40
Sugar,	1.5	6
Molasses (dry matter only),	0.75	3
	25.00	100

Calorimetric tests of molasses, sugar and fibre indicate the following values: —

Molasses	6,956 B. T. U.
Sugar	7,223 "
Fibre	8,325 "

In Table No. 28 the total heat values are based upon those of the constituents given above, and a fair sample of Pennsylvania coal having a heat value of 14,000 B.T.U. is taken as the basis of comparison with coal.

Table No. 28. — Value of One Pound of Mill Bagasse at Different Extractions upon Cane of 10 per cent Fibre and Juice of 15 per cent Total Solids.

Per cent Extraction on Weight of Cane.	Per cent Moisture, in Bagasse.	FIBRE.		SUGAR.		MOLASSES.		Total Heat developed. B. T. U.	Heat required to evaporate the Water present. B. T. U.	Heat Available. B. T. U.	Pounds Bagasse required to equal 1 lb. Coal of 14,000 B. T. U. Calorific Power.	Coal equivalent per Ton of Cane. Pounds.	Temperature of Fire. Fahr.
		Per cent, in Bagasse.	Fuel Value. B. T. U.	Per cent, in Bagasse.	Fuel Value. B. T. U.	Per cent, in Bagasse.	Fuel Value. B. T. U.						
90	0.00	100.00	8,325	—	—	—	—	8,325	—	8,325	1.68	119	2,465°
85	28.33	66.67	5,550	3.33	240	1.67	116	5,900	339	5,561	2.52	119	2,236
80	42.50	50.00	4,162	5.00	361	2.50	174	4,697	509	4,188	3.34	120	2,023
75	51.00	40.00	3,330	6.00	433	3.00	209	3,972	611	3,361	4.17	120	1,862
70	56.67	33.33	2,775	6.67	482	3.33	232	3,489	679	2,810	4.98	120	1,732
65	60.71	28.57	2,378	7.15	516	3.57	248	3,142	727	2,415	5.80	121	1,612
60	63.75	25.00	2,081	7.50	541	3.75	261	2,883	764	2,119	6.61	121	1,513
55	66.12	22.22	1,850	7.78	562	3.88	270	2,682	792	1,890	7.40	121	1,427
50	68.00	20.00	1,665	8.00	578	4.00	278	2,521	815	1,706	8.21	122	1,350
45	69.55	18.18	1,513	8.18	591	4.09	284	2,358	833	1,555	9.00	122	1,284
40	70.83	16.67	1,388	8.33	601	4.17	290	2,279	849	1,430	9.79	123	1,222
25	73.67	13.33	1,110	8.67	626	4.33	301	2,037	883	1,154	12.13	124	1,077
15	75.00	11.77	980	8.82	637	4.41	307	1,924	899	1,025	13.66	124	1,002
0	76.50	10.00	832	9.00	650	4.50	313	1,795	916	879	15.93	126	906

In somewhat abbreviated form the fuel values of one pound of diffusion bagasse, at various degrees of moisture, are given in Table No. 29.

Table No. 29. — Fuel Values of One Pound of Diffusion Bagasse at Various Degrees of Moisture.

Moisture in Bagasse. Per cent.	Heat developed per Pound of Bagasse. B. T. U.	Heat Available per Pound of Bagasse. B. T. U.	Number of Pounds of Bagasse Equivalent to 1 lb. of Coal.	Estimated Temperature of Fire. Fahr.
0	8,325	8,325	1.68	2,465°
20	6,660	6,420	2.18	2,294
30	5,827	5,468	2.56	2,186
40	4,995	4,516	3.10	2,049
50	4,162	3,563	3.93	1,870
60	3,330	2,611	5.41	1,627
70	2,497	1,658	8.44	1,281
75	2,081	1,183	11.90	1,045

Peat. — Intermediate between wood and coal may be placed peat, which is the result of one of the most important geological changes now in progress. In certain swampy regions in the temperate latitudes there occur immense quantities of semi-aquatic plants, which, under special conditions of heat and moisture, are undergoing a curious chemical transformation, whereby the oxygen of the plant is eliminated, leaving behind as peat a spongy carbonaceous residue. This is found in beds varying from 1 or 2 to 40 feet in depth. That near the surface, which is in a less advanced state of decomposition, is light, spongy and fibrous, of yellow or light reddish-brown color; lower down it is more compact, and darker in color; while in the lowest strata the color is almost black, and the peat is pitchy and unctuous, with scarcely any evidence of the fibrous texture which exists in the higher strata and in the original vegetable matter from which it was formed.

In its natural condition, peat generally contains from 75 to 80 per cent of its entire weight of water, occasionally amounting to 85 or even 90 per cent. Evidently it is thus totally unfitted for use as a fuel until it has been dried. By the process of drying it shrinks very decidedly, its specific gravity, when dry, varying from 0.22 or 0.34 for the surface peat, which is light and porous, to 1.06 for the lowest peat, which is very dense. Owing to the abundance of other fuels, peat has been but little used in this country; but in Ireland, Germany and Sweden it has already found an extensive field, not only in domestic but in metallurgical operations.

The composition of ordinary Irish peats, both exclusive and inclusive of the moisture, which they always contain in their natural condition, is displayed herewith in Table No. 30.

Table No. 30.—Composition of Irish Peats.

DESCRIPTION.	Moisture.	Carbon.	Hydrogen	Oxygen.	Nitrogen.	Sulphur.	Ash.	Coke.
EXCLUSIVE OF MOISTURE.								
Good air-dried,	——	59.7	6.0	31.9		——	2.4	——
Poor air-dried,	——	59.6	4.3	29.8		——	6.3	——
Dense, from Galway,	——	59.5	7.2	24.8	2.3	0.8	5.4	44.3
Averages,	——	59.6	5.8	29.6		0.3	4.7	——
INCLUSIVE OF MOISTURE.								
Good air-dried,	24.2	45.3	4.6	24.1		——	1.8	——
Poor air-dried,	29.4	42.1	3.1	21.0		——	4.4	——
Dense, from Galway,	29.3	42.0	5.1	17.5	1.7	0.6	3.8	31.3
Averages,	27.8	43.1	4.3	21.4		0.2	3.3	——

The average composition of Irish peat, disregarding sulphur, which is seldom present, at least in quantity sufficient to have any appreciable influence, may be taken to be as given in Table No. 31.

Table No. 31.—Average Composition of Irish Peat.

CONSTITUENTS.	Perfectly Dry.	Including 25 per cent of Moisture.	Including 30 per cent of Moisture.
Carbon,	59.0	44.0	41.2
Hydrogen,	6.0	4.5	4.2
Oxygen,	30.0	22.5	21.0
Nitrogen,	1.25	1.0	0.8
Ash,	4.0	3.0	2.8
Moisture,	——	25.0	30.0

By calculation the thermal value of dry Irish peat of the average composition, shown in Table No. 31, is—

Carbon . . . . $14,650 \times 0.59 = 8,643.5$ B. T. U.

Hydrogen . . $62,100\,(.06 - \frac{30}{8}) = 1,397.25$ B. T. U.

Total B. T. U. . . . 10,040.75

Coal. — The extensive distribution, the portable character and the heat value of coal make it the principal fuel of all civilized nations. Coal is in effect the reservoir of the stored energy of the sun, by the action of whose heat rays it was produced. It is a fossil fuel for whose existence geology thus accounts: During that period of the earth's formation known as the carboniferous age, vegetation was rank in the extreme. The atmosphere contained an amount of carbonic acid far in excess of that now present. The presence in the atmosphere of this excess of carbon, which is the food of the plants, as well as the temperature and the climatic conditions, were all favorable to the most prolific development of plant life. Age after age was employed by this vegetable growth in freeing the atmosphere from carbonic acid, and in storing up the potential energy of the sunlight as woody fibre in the form of carbon, separated from oxygen. By this continuous process of growth and death of vegetable matter the earth became strewed with the remains, which were gradually compacted into peat beds of enormous extent. With succeeding climatic and geological changes, these peat beds, one after another, become submerged and overlaid by thousands of feet of sandstone, limestone and slate. Under the tremendous pressure thus exerted the peat beds were compressed and converted by successive stages into lignite, brown coal, gaseous coal, bituminous coal and semi-anthracites. In certain localities, through some igneous agency, the coal beds were subjected to distillation, the hydro-carbons in the forms of oil or gas being thus driven away to be stored in the natural reservoirs of the earth, while anthracite, shiny and rock-like, was left behind.

It must be obvious that sharp lines of demarcation between the various kinds of coal cannot exist, and, therefore, that they can only be approximately classified, for one form merges into another. A fair illustration of the different stages in the process of alteration of wood fibre into anthracite coal is presented in Table No. 32.

Table No. 32. — Conversion of Wood Fibre into Anthracite.

Description.	Carbon.	Hydrogen.	Oxygen.
Wood fibre (cellulose),	52.65	5.25	42.10
Peat,	60.44	5.96	33.60
Lignite,	66.96	5.27	27.76
Lignite (brown coal),	74.20	5.89	19.90
Coal (bituminous),	76.18	5.64	18.07
Coal (semi-anthracite),	90.50	5.05	4.40
Anthracite,	92.85	3.96	3.19

Coals are usually classified according to the amounts of carbon and volatile matter which are present in their composition, although different methods are adopted by different authorities. The following is the classification generally adopted, beginning with those containing the greatest proportion of carbon:—

Anthracites		Hard anthracites. Semi or gaseous anthracites.
Common bituminous coal,	Semi-bituminous coals	Semi-bituminous cherry coal. Semi-bituminous splint coal.
	Bituminous coal	Caking coal. Cherry coal. Splint coal.
Hydrogenous or gas coal		Cannel coal. Hydrogenous shaly coal. Asphaltic coal.
Lignite.		

The general composition of these coals has already been given in Table No. 13. In the consideration of their characteristics they will be taken up in the order of their geological formation. Their progressive alteration from wood to coal is thus clearly indicated.

Lignite.—Although classed among mineral coals, from a geological standpoint, lignite properly occupies a position between peat and bituminous coal. It is believed to be of later origin than bituminous coal, and is in a less advanced stage of decomposition. The woody fibre and vegetable texture of lignite are almost entirely wanting in coal, although there is little question as to their common origin. Although much like brown coal in general appearance, lignite differs from it in the fact that upon distillation it yields acetic acid, while brown coal produces only ammoniacal liquor. Like peat, lignite presents much variety in appearance, some specimens being almost as hard as true coal, while others possess a distinctly woody structure and are of a light-brown color. It has an uneven fracture and a dull and somewhat fatty lustre. Being easily broken, it will not readily bear transportation, while exposure to the weather causes it to rapidly absorb moisture and to crumble easily. Its value as a fuel is, therefore, limited, for it must be used near its place of occurrence, and very soon after it is mined. It is non-caking and yields but moderate heat, being inferior to even the poorer varieties of bituminous coal. In this country its use is decidedly limited, being restricted to the locality of the mines which produce it. It is plenteous, however, west of the Mississippi, in which territory it is used to a considerable extent. The three analyses which are presented in Table No. 33 give the average composition of samples from the widely separated states of Kentucky, Washington and Colorado.

Table No. 33. — Composition of Lignite.

LOCALITY.	Specific Gravity.	Fixed Carbon.	Volatile Combustible Matter	Water.	Ash.	Total Volatile Matter.	Coke.
Kentucky,	1.201	40.0	23.0	30.0	7.0	53.0	47.0
Washington,		52.85	31.75	7.00	3.00	61.25	38.75
Colorado,	1.271	41.25	46.00	3.50	9.25	50.50	49.50

Bituminous Coal. — The classification of bituminous coal is rendered difficult because of the lack of definite lines of demarcation between the varieties. As a rule, however, coal containing as much as 18 to 20 per cent of volatile combustible is called bituminous. Some bituminous coal yields, upon analysis, as much as 50 per cent of volatile matter and sometimes more. In *proximate composition*,— namely, in fixed carbon, volatile matter and earthy matter,— the bituminous coals may be regarded as ranging between the following general limits: —

Fixed carbon	52 to 84	per cent.
Volatile matter	12 to 48	“
Earthy matter	2 to 20	“
Sulphur	1 to 3	“

The amount of water expelled by heating to 212° is from 1 to 4 per cent.

In *ultimate composition*, as shown by refined analysis, the approximate range of composition is as follows: —

Carbon	75 to 80	per cent.
Hydrogen	5 to 6	“
Nitrogen	1 to 2	“
Oxygen	4 to 10	“
Sulphur	0.4 to 3	“
Ash	3 to 10	“

In its external properties, ordinary bituminous coal varies in color from a pitch black to a dark brown, with a lustre that is vitreous or resinous in the more compact specimens, and silky in those showing traces of vegetable fibres. Irrespective of natural joints, the fracture of bituminous coal is generally conchoidal. The distinctive characteristic of this fuel is the emission of yellow flame and smoke when burning.

All bituminous coals may be classified on broad lines as either caking or non-caking.

Caking Coal is the name given to any coal which, when heated, seems to fuse together and swell in size, becomes pasty in appearance and emits a sticky substance over the surface, while liberating small streams of gas which burn with a bright yellow or reddish flame terminating in smoke. It is characteristic of caking coal that the pasty lumps will cohere in the fire and form spongy-looking masses, not infrequently covering the entire surface of the grate. Such coals, unusually rich in volatile hydro-carbons, are considered most valuable for gas manufacture.

Non-Caking Coal has the property of burning freely in the fire; hence the common appellation, "free-burning coal." The heat does not cause the lumps to fuse or run together. The block coal of the Western States is a representative non-caking coal. It consists of successive layers which are easily separated into thin slices. The surfaces which are thus displayed are generally covered with a layer of very finely divided fibrous carbon and are dull and lustreless. When coal of this character is broken at right angles to this lamination the surface is bright and glistening.

The ultimate composition of various bituminous coals is given in Table No. 13, which has already been presented; while among the coals listed in Table No. 36, which follows on succeeding pages, are also many that would be classed as bituminous.

Cannel Coal is a variety of bituminous coal very rich in carbon. It kindles readily, burns without melting and emits a bright flame like that of a candle. It differs greatly in appearance from all other bituminous coals, being very homogeneous, having a dull, resinous lustre, and breaking without following any distinct line of fracture. It is exceedingly valuable as a gas coal because of its richness in hydro-carbons, but is little used in this country as a steam or boiler coal. The proximate analysis of a few typical American specimens is presented in Table No. 34.

Table No. 34. — Composition of Cannel Coal.

LOCALITY.	Specific Gravity.	Fixed Carbon.	Volatile Matter.	Earthy Matter.
Franklin, Pa.,		40.13	44.85	15.02
Dorton's Branch, Ky.,	1.25	55.1	42.9	2.0
Breckenridge, Ky.,		32.0	55.7	12.3
Davis County, Ind.,	1.23	42.0	52.0	6.0

Semi-Bituminous Coal is softer and contains more volatile matter than true anthracite coal, but in its general characteristics closely approaches that fuel. It resembles in appearance the anthracites more closely than it does the bituminous coals, but its fracture is less conchoidal than that of the former; it is lighter and both kindles and burns more rapidly. Because of this latter feature it is extremely valuable as a fuel, for when burned it readily gives off a great quantity of heat and can always be relied upon to keep up an intense and free-burning fire requiring comparatively little attention, readily cleaned and kept in good condition. It is, when pure, almost entirely free from smoke and soot.

The proximate analysis of semi-bituminous coal from Cumberland, Md., and Blossburg, Pa., is given in Table No. 35.

Table No. 35.—Composition of Semi-Bituminous Coal.

LOCALITY.	Specific Gravity.	Fixed Carbon.	Volatile Matter.	Sulphur.	Earthy Matter.
Cumberland, Md.,	1.41	68.44	17.28	0.71	13.98
Blossburg, Pa.,	1.32	73.11	15.27	0.85	10.77

Semi-Anthracite Coal.—Among the semi-anthracite coals are classed those which contain from 7 to 8 per cent of volatile combustible matter. Because of the presence of this ingredient, which apparently exists in the gaseous state in the cells or cracks of the coal, this variety kindles more readily and burns more rapidly than hard anthracite. Analysis of Wilkesbarre, Pa., semi-anthracite, which is compact, conchoidal, iron black and shiny, shows the following to be its composition:—

Fixed carbon	88.90 per cent.
Volatile matter	7.68 "
Earthy matter	3.49 "
	100.07

Its specific gravity is 1.4.

Anthracite Coal.—Pure anthracite, sometimes called blind coal, ignites slowly, is a poor conductor of heat, and burns at a very high temperature. When pure, it consists of—

Carbon	90 to 94 per cent.
Hydrogen	1 to 3 "
Oxygen and nitrogen	1 to 3 "
Water	1 to 2 "
Ash	3 to 4 "

It is thus evident that it is composed almost entirely of carbon; in fact, this is its distinguishing characteristic. The hydro-carbons, as evidenced in the volatile constituents, are present in very small proportion. As a consequence, it is not a long-flaming coal, but when in a state of incandescence its radiant power is great, owing to the intensity of combustion of the practically pure carbon of which it consists.

In the process of burning it neither swells, softens nor gives off smoke. The flame is quite short, of a yellowish tinge, changing to a faint blue, and largely due to the presence of water which is decomposed by the heat. This flame is free from particles of solid carbon, and has the appearance of being transparent. Anthracite coal is homogeneous in structure; its fracture is decidedly conchoidal, and it is but slightly affected by exposure to the weather.

Analysis of anthracite coal from Tamaqua, Pa., shows it to consist of —

Carbon	92.07	per cent.
Volatile matter	5.03	"
Earthy matter	2.90	"
	100.00	

and to have a specific gravity of 1.57.

Geographical Classification. — Although widely distributed throughout the United States, the various kinds of coal may be geographically classified in a general manner, as follows: —

Anthracite		Eastern portion of Allegheny Mountains and Rocky Mountains of Colorado.
Bituminous coals,	Caking,	Mississippi Valley.
	Non-caking,	Maryland and Virginia.
	Cannel,	Pennsylvania, Indiana and Missouri.
Lignites		Colorado, Kentucky and Washington.

A carefully selected list of analyses of representative American coals, geographically arranged, is presented in Table No. 36[1], which makes clear the differences which exist even in coals from the same locality. The ultimate value of any coal as a steam producer must be measured by the amount of water it can evaporate when properly burned in the furnace of a steam boiler. But this value may be modified by certain characteristics of the coal; and, as will be pointed out later, the efficiency of a fuel is to a considerable extent dependent upon the character of the boiler and furnace in connection with which it is consumed.

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[1] Helios. E. D. Meier. St. Louis, 1895.

Table No. 36.—Composition and Fuel Value of American Coals.

COAL, NAME OR LOCALITY.	Constituents in per cent of Total Weight.					Fuel Value per Pound of Coal.		
	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	B. T. U. Calculated.	B. T. U. by Calorimeter.	Theoretical Evaporation in lbs. from and at 212°.
1	2	3	4	5	6	7	8	9
ARKANSAS.								
Coal Hill, Johnson Co.,	1.35	14.93	74.06	9.66	3.04	13,713		14.1
Coal Hill, Johnson Co.,	1.70	14.60	74.91	8.79	3.04		11,812	12.22
Huntington Co.,	1.30	18.95	71.51	8.24	0.78		11,756	12.17
Huntington Co.,	1.30	18.90	73.15	6.65	0.75		11,907	12.32
Huntington Co.,	1.27	18.89	71.74	8.10	0.65		12,537	12.97
Lignite,						9,215		9.54
Jenny Lind, Sebastian Co.,	1.26	17.64	72.48	8.62	2.11	13,964		14.4
Spadra, Johnson Co.,	1.47	13.27	78.63	6.63	1.60	14,420		14.9
COLORADO.								
Lignite,						13,560		14.04
Lignite,						13,865		14.35
Lignite, slack,	14.80	32.00	42.86	10.34	0.76	8,500		8.80
Lignite, slack, North Colorado,	18.88	31.74	40.08	9.30	0.61			
Rouse Mine,	3.13	37.32	30.00	8.25				
ILLINOIS.								
Big Muddy, Jackson Co.,	7.39	28.28	53.87	10.46	0.98		11,466	11.87
Big Muddy, Jackson Co.,	6.12	30.95	53.74	9.19	1.22		11,529	11.93
Big Muddy, Jackson Co.,	5.85	31.84	55.72	6.59	2.92		11,781	12.19
Big Muddy, Jackson Co.,	6.35	31.50	55.25	6.90	2.02	12,567		13.0
Bureau Co.,						13,025		13.48
Colchester,	11.60	25.02	44.76	18.62			9,848	10.19
Colchester, slack,	5.30	25.45	38.15	31.10	1.20		9,035	9.35
Collinsville, Madison Co.,	9.20	45.89	31.57	13.34	5.34		10,143	10.50
Dumferline, slack,	9.64	28.86	39.48	22.02			9,401	9.73
Duquoin Jupiter, Perry Co.,	11.30	30.31	49.91	8.48	0.91		10,710	11.08
Ellsworth, Macoupin Co.,	9.26	42.22	42.17	6.35	2.62	12,175		12.60
Gillespie, Macoupin Co.,	12.61	30.58	45.27	11.54	1.45		9,739	10.09
Girard, Macoupin Co.,	9.70	34.39	45.76	10.15	3.49		9,954	10.30
Girard, Macoupin Co.,	8.90	32.25	42.89	15.96	8.10		10,269	10.63
Heitz Bluff, St. Clair Co.,	8.95	37.81	48.24	5.00	3.27		10,332	10.69
Johnson's, St. Clair Co.,	5.50	40.14	40.53	13.83	4.80	11,723		12.10
Loose's, Sangamon Co.,	10.71	37.62	45.07	6.60	2.39	11,479		11.9
Mercer Co.,						13,123		13.58

COAL. NAME OR LOCALITY. 1	Constituents in per cent of Total Weight.					Fuel Value per Pound of Coal.		
	Moisture. 2	Volatile Matter. 3	Fixed Carbon. 4	Ash. 5	Sulphur. 6	B. T. U. Calculated. 7	B. T. U. by Calorimeter. 8	Theoretical Evaporation in lbs. from and at 212°. 9
ILLINOIS.—Concluded.								
Montauk Co.,						12,659		13.10
Mt. Olive, Macoupin Co.,	10.38	36.68	46.10	6.84	3.53	11,763		12.2
Oakland, St. Clair Co.,	8.30	34.40	43.12	14.18	4.42		10,395	10.76
Reinecke, St. Clair Co.,	7.56	39.81	42.49	10.14	4.02	11,720		12.1
Riverton, Sangamon Co.,	11.06	37.94	42.98	8.02	3.27	11,406		11.8
St. Clair,	7.80	30.69	39.68	21.83	9.62		9,261	9.58
St. Clair,	10.25	33.10	41.79	14.86	6.92		10,294	10.65
St. Clair,	11.15	34.19	44.94	9.72	4.27		10,647	11.02
St. Bernard,	14.36	30.86	48.39	6.39	1.38		10,080	10.44
St. John, Perry Co.,	9.82	28.35	45.77	16.08	2.06		9,765	10.10
St. John, Perry Co.,	13.60	24.46	43.54	15.40	1.83		9,828	10.18
Streator, La Salle Co.,	12.01	35.32	48.78	3.89	2.38		11,403	11.80
Trenton, Clinton,	13.34	30.39	51.96	4.31	0.92		10,584	10.96
Trenton, Clinton,	9.95	31.04	52.03	6.98	1.04		11,245	11.63
Vulcan Nut, St. Clair Co.,	7.44	30.86	45.09	16.61	1.32		9,450	9.78
Vulcan Nut, St. Clair Co.,	10.30	27.91	48.99	12.80	0.71		10,626	11.00
INDIANA.								
Block,	3.50	32.50	63.00	1.00	0.98	14,020		14.5
Block,						13,588		14.38
Caking,						14,146		14.64
Cannel,						13,097		13.56
INDIAN TERRITORY.								
Atoka,	6.66	35.42	51.32	6.60	3.73		11,088	11.47
Choctaw Nation,	1.59	23.31	66.85	8.25	1.18		12,789	13.23
IOWA.								
Good Cheer,	10.85	30.32	31.38	27.45	7.32		8,702	9.01
KENTUCKY.								
Caking,						14,391		14.89
Cannel,						15,198		16.76
Cannel,						13,360		13.84
Lignite,						9,326		9.65
MISSOURI.								
Bevier Mines,							9,890	10.24

COAL, NAME OR LOCALITY. 1	Constituents in per cent of Total Weight.					Fuel Value per Pound of Coal.		
	Moisture. 2	Volatile Matter. 3	Fixed Carbon. 4	Ash. 5	Sulphur. 6	B. T. U. Calculated. 7	B. T. U. by Calorimeter. 8	Theoretical Evaporation in lbs. from and at 212°. 9
MARYLAND.								
Cumberland,						12,226		12.65
George's Creek,						13,500		13.98
NEW MEXICO.								
Coal,	2.35	35.53	50.24	11.88	0.61		11,756	12.17
OHIO.								
Briar Hill, Mahoning Co.,	2.47	31.83	64.25	1.45	0.56	13,714		14.2
Hocking Valley,	8.25	35.88	53.15	2.72	0.43	13,414		13.9
PENNSYLVANIA.								
Anthracite,						14,199		14.70
Anthracite,						13,535		14.01
Anthracite,						14,221		14.72
Anthracite, pea,	2.04	6.36	78.41	13.19		12,300		12.73
Anthracite, buckwheat,	3.88	3.84	81.32	10.96	0.67	12,200		12.63
Cannel,						13,143		13.60
Connellsville,						13,368		13.84
Pittsburgh (average),	1.80	35.34	54.94	7.92	1.97		13,104	13.46
Pittsburgh, coking,	1.43	30.22	61.87	6.48	1.35	14,415		14.9
Youghiogheney,	1.96	34.06	58.98	5.00			12,936	13.39
Youghiogheney,	2.02	32.14	58.96	6.88	0.88		12,600	13.03
Reynoldsville,	1.20	27.12	65.88	5.80			12,981	13.44
TENNESSEE.								
Glen Mary, Scott Co.,	2.15	31.47	61.63	4.75	0.94		13,167	13.63
TEXAS.								
Fort Worth,	14.42	30.03	42.53	13.02	1.47		9,450	9.78
Fort Worth,	4.60	34.72	49.27	11.41	1.56		11,403	11.80
Lignite,						12,962		13.41
WEST VIRGINIA.								
New River,						14,200		14.70
New River,						13,400		13.87
New River,	0.94	18.19	75.89	4.68	0.30			
New River,	0.76	18.65	79.26	1.11	0.23			

Petroleum. — The only natural liquid fuel is crude petroleum oil. This is distinctly a hydro-carbon liquid, and is found in abundance in certain localities in America and Europe. The principal sources of supply are, however, in the Ohio Valley of the United States, and on the borders of the Caspian Sea in Eastern Europe. As already explained, it appears to be the result of distillation of coal under great pressure. It is found in natural cavities beneath the earth's surface, whence it is either pumped, or flows to the surface after the manner of operation of an artesian well.

Crude petroleum is dark brown in color, with a perceptible greenish tinge, and has a specific gravity which averages about 0.8. It is composed of a great number of liquid hydro-carbons varying widely in specific gravity and chemical composition, and each separable from the others by fractional distillation. The ultimate analysis of an average sample indicates about the following composition: —

Carbon	84 per cent.
Hydrogen	14 "
Oxygen	2 "
	100 per cent.

Allowing for the combination of the inherent oxygen with its equivalent of hydrogen to form water, the practical composition becomes —

Carbon	84 per cent.
Hydrogen	13.75 "
Water	2.25 "
	100 per cent.

The heat value of a pound of petroleum of the above composition is, therefore, —

Carbon . . . 0.84 × 14,650 = 12,306 B. T. U.
Hydrogen . . 0.1375 × 62,100 = 8,539 "
20,845 B. T. U.

Natural Gas. — Although springs of natural gas exist in nearly every State of the Union, the commercial use of this fuel is practically limited to the States of Indiana, New York, Pennsylvania and Ohio, in which the largest supply is to be found. This gas is an inseparable companion of natural oil or petroleum, as would be the natural consequence of their both being the products of distillation of coal. In its composition natural gas varies greatly. Not only is there a marked difference in composition between the gas from different wells, but also between samples which are taken at different times from the same well.

The variation in the composition of natural gas is illustrated by the results of six analyses, made within a period of three months, of different samples from a well near Pittsburg, Pa., as presented in Table No. 37.

Table No. 37.—Variation in Composition of Natural Gas.

CONSTITUENTS.	1	2	3	4	5	6
Marsh gas,	57.85	75.16	72.18	65.25	60.70	49.58
Hydrogen,	9.64	14.45	20.02	26.16	29.03	35.92
Ethylic hydride,	5.20	4.80	3.60	5.50	7.92	12.30
Olefiant gas,	0.80	0.60	0.70	0.80	0.98	0.60
Oxygen,	2.10	1.20	1.10	0.80	0.78	0.80
Carbonic oxide,	1.00	0.30	1.00	0.80	0.58	0.40
Carbonic acid,	0.00	0.30	0.80	0.60	0.00	0.40
Nitrogen,	23.41	2.89	0.00	0.00	0.00	0.00

Analyses from various wells in Indiana and Ohio indicate the composition to be as given in Table No. 38.

Table No. 38.—Composition of Natural Gas from Ohio and Indiana.

CONSTITUENTS.	OHIO.			INDIANA.			
	Fostoria.	Findlay.	St. Mary's	Muncie.	Anderson	Kokomo.	Marion.
Hydrogen,	1.89	1.64	1.94	2.35	1.86	1.42	1.20
Marsh gas,	92.84	93.35	93.85	92.67	93.07	94.16	93.57
Olefiant gas,	0.20	0.35	0.20	0.25	0.47	0.30	0.15
Carbonic oxide,	0.55	0.41	0.44	0.45	0.73	0.55	0.60
Carbonic acid,	0.20	0.25	0.23	0.25	0.26	0.29	0.30
Oxygen,	0.35	0.39	0.35	0.35	0.42	0.30	0.55
Nitrogen,	3.82	3.41	2.98	3.53	3.02	2.80	3.42
Hydrogen sulphide,	0.15	0.20	0.21	0.15	0.15	0.18	0.20

Artificial Fuels. — Although artificial fuels serve a useful purpose in steam-making, their use is by no means as extended as that of the natural fuels. The desirability of employing one in preference to the other is dependent largely upon financial considerations; and that fuel is to be chosen which, other things equal, will evaporate the most water for a given total expenditure. Artificial fuels may be broadly classified under the headings charcoal, coke, fuel gas and patent fuels.

Charcoal. — Wood, protected from the atmosphere and heated at about 600°, gives up its gaseous or volatile elements, and is converted into charcoal. The best charcoal consists almost entirely of pure carbon; but in so far as the process of manufacture falls short of perfection, so the proportion of carbon is reduced. Charcoal is distinctly the result of a process of carbonization; and under the condition of distillation in vessels externally heated to various temperatures, its quality is improved as its temperature is increased. The results of this process when applied to black alder, previously dried at about 300°, are presented in Table No. 39.

Table No. 39. — Composition of Carbon Produced at Various Temperatures.

Temperature of Carbonization.	CONSTITUENTS OF THE SOLID PRODUCT.				
	Carbon.	Hydrogen.	Oxygen.	Nitrogen and Loss.	Ash.
302° F.	47.51	6.12	46.29	0.08	47.51
392	51.82	3.99	43.98	0.23	39.88
482	65.59	4.81	28.97	0.63	32.98
592	73.24	4.25	21.96	0.57	24.61
662	76.64	4.14	18.44	0.61	22.42
810	81.64	4.96	15.24	1.61	15.40
1,873	81.97	2.30	14.15	1.60	15.30

Peat charcoal, produced by the carbonization of ordinary air-dried peat, is very friable and porous, and extremely difficult to handle without reducing it to very small particles almost powdery in their character. Although it is easily ignited and burns readily, its physical characteristics are such as to prevent its general use.

Coke. — The residual product of the carbonization of bituminous coal is known as coke. By this process the hydro-carbon gases are expelled, and the coal is reduced to a substance somewhat porous in its character and consisting almost entirely of carbon. The coke produced by the partial combustion of coal in coke ovens is dark gray in color, hard, porous and brittle, with a slightly metallic lustre. That resulting as a by-product from the distillation of gas in the retorts of gas works is not so hard, ignites more readily and burns with a draft less intense than that required for the combustion of coke which has been formed by the first method. It is, therefore, better adapted as a fuel for steam-boiler furnaces.

The quality of such coke is affected by the temperature and by the pressure under which distillation takes place. From experiments by Mr. A. L. Steavenson, reported to the Iron and Steel Institute at Newcastle, Eng., it appeared that with a furnace of special construction and coal of the following composition, —

Oxygen	6.7	per cent.
Carbon	84.9	"
Hydrogen	4.5	"
Nitrogen	1.0	"
Sulphur	0.6	"
Ash	2.3	"

the yield was about 60 per cent of coke of the following composition: —

Carbon	96.2	per cent.
Ash	3.8	"

Hence the composition and relative weight of the materials lost in coking were —

Carbon	68.1	per cent.
Hydrogen	11.2	"
Nitrogen	2.5	"
Sulphur	1.6	"
Oxygen	16.6	"

Fuel Gas. — Although carbonic oxide and hydrogen are combustibles, the production of which is incident to the combustion of all fuels, they are never independently manufactured for use as fuels. But hydrogen and carbon, associated in the form of volatile hydro-carbons, serve most excellently the purposes of fuels, although their cost must determine their efficiency. Notwithstanding the fact that illuminating gas made from bituminous coal by distillation in retorts has been in common use for nearly a century, the idea of directly converting a solid fuel into one of gaseous form for its readier utilization for producing heat has only been carried into practice during the past twenty-five or thirty years. Gas as a fuel first appeared in the form of "producer" gas, and was primarily introduced for metallurgical purposes. In its manufacture, air, mixed with water vapor, was passed, under powerful pressure, through a thick bed of burning coal. As a result the coal was only burned to carbonic oxide, while the watery vapor was decomposed so that the resulting gas from the producer was a mixture of about one-half nitrogen and one-fourth carbonic oxide, with varying proportions of hydrogen and hydro-carbons. Its composition depends upon the proportions of the elements in the original fuel. This process, however, inherently consumes about one-third of the total calorific value of the fuel, thereby reducing by that amount the resultant heating power.

Water gas is the result of a somewhat similar process, which differs principally from that employed in the manufacture of producer gas in that it is intermittent, first air and then steam being forced through a bed of incandescent fuel. For illuminating purposes this gas is carburetted, so that it actually exceeds by volume the value of coal gas.

Including natural gas the relative volumes and weights of gaseous fuels are: —

	By Weight.	By Volume.
Natural gas,	1,000	1,000
Coal gas,	949	666
Water gas,	292	292
Producer gas,	76.5	130

By weight and volume the composition of these gases is given in Table No. 40.

Table No. 40. — Composition of Fuel Gases.

Constituents.	By Volume.				By Weight.			
	Natural Gas.	Coal Gas.	Water Gas.	Producer Gas.	Natural Gas.	Coal Gas.	Water Gas.	Producer Gas.
Hydrogen,	2.18	46.0	45.0	6.0	0.268	8.21	5.431	0.458
Marsh gas,	92.60	40.0	2.0	3.0	90.383	57.20	1.931	1.831
Carbonic oxide,	0.50	6.0	45.0	23.5	0.857	15.02	76.041	25.095
Olefiant gas,	0.31	4.0	0.0	0.0	0.531	10.01	0.000	0.000
Carbonic acid,	0.26	0.5	4.0	1.5	0.700	1.97	10.622	2.517
Nitrogen,	3.61	1.5	2.0	65.0	6.178	3.75	3.380	69.413
Oxygen,	0.34	0.5	0.5	0.0	0.666	1.43	0.965	0.000
Watery vapor,	0.00	1.5	1.5	1.0	0.000	2.41	1.630	0.686
Sulphydric acid,	0.20	——	——	——	0.417	——	——	——

In this table the natural gas was from Findlay, Ohio, the coal gas was probably an average sample purified for illuminating purposes, the water gas was made for heating and consequently unpurified, and the producer gas was made from anthracite at the Pennsylvania Steel Works.

Patent Fuels. — Under this title may be classed a large variety of prepared fuels, consisting in the main of the particles of some finely divided combustible pressed and cemented together by a substance possessing the necessary adhesive and inflammable properties.

In the process of coal mining, sorting and shipping, a considerable amount is broken into fragments, too small for ordinary commercial use. This refuse, commonly denominated "culm," possesses practically the same calorific value as the coal of which it originally formed a part; but its finely divided character is not conducive to its successful use in an ordinary boiler furnace. It may, however, by special machinery, be mixed with sufficient pitch or coal tar, and moulded into lumps of desirable size.

In this country the relative price of coal is so low, as compared with the cost of manufacture of such pressed fuel, that the financial return hardly warrants the attempt to thus utilize the culm. In France, and some other European countries, fuel of this character, in the form of "briquettes," is regularly made of coal dust,—bituminous and semi-anthracite,—and quite extensively used. To some extent the slow progress made in the manufacture of briquettes in this country is doubtless due to the imperfect systems of washing and jigging which are necessary to reduce the percentage of ash, which never ought to exceed 10 per cent in such fuel.

The attempt has also been made, with varying success, according to the conditions, to feed the coal in the form of dust directly to the boilers, by forcing it into a strong air current, which thus spreads it throughout the furnace, while at the same time furnishing the oxygen necessary for combustion.

By means of glue, tar, pitch, resin and the like, sawdust, charcoal, peat, tan and similar refuse have been cemented together for use as a fuel. But, except in comparatively few instances, the cost of manufacture of prepared fuels has, in this country at least, rendered them but little, if any, more economical than coal.

CHAPTER V.

EFFICIENCY OF FUELS.

Measure of Efficiency. — The ultimate efficiency of a fuel should be expressed in the total amount of heat it is capable of generating. The proportion of that heat which is utilized depends upon the efficiency of the boiler or other heat-abstracting device. Commercially, however, the heat value of fuels is generally measured relatively to each other, and expressed in the number of pounds of water evaporated per pound of fuel. In practice, the physical character of the fuel, the form and construction of the boiler and furnace, the amount of air supplied, and other conditions, have an important influence upon the attainable results. In fact, the effect of these variables is such as to render an accurate comparison of fuels a difficult matter.

In their ultimate efficiency they may, however, be considered relatively to each other or to an established standard. As carbon is the most important element in the composition of all fuels, it may reasonably be selected as such a standard. For the purposes of comparison, Table No. 41 has been prepared to show the efficiency of fuels as measured in thermal units, determined by analysis or calorimetric test when compared with pure carbon as a standard, having a thermal value of 14,650 B. T. U. Measured by this standard, a coal having very little ash and a large amount of hydrogen may, because of its extremely high heating power, actually show an efficiency above 100 per cent when compared with carbon. As the theoretical efficiency of a fuel can never be realized in practice, there have been incorporated in this table the fuel efficiency and the number of pounds of water evaporated from and at 212° per pound of fuel at various boiler efficiencies.

Evidently, the actual efficiency of a given fuel is here, as in all cases, dependent upon the efficiency of the boiler. It is obvious, however, that an evaporation of 15.2 pounds of water from and at 212° per pound of best coal represents an ideally perfect result, with 100 per cent efficiency of both fuel and boiler, unless the fuel contains sufficient volatile matter to raise its total heat above 14,650 B. T. U.

For the purposes of practical comparison there is usually determined the number of pounds of water of a given temperature that can be evaporated into steam of a given pressure by the combustion of one pound of the fuel. This is reduced for direct comparison to the standard of temperature of water at 212°, and steam of atmospheric pressure; namely, of a temperature of 212°. Under these conditions, as no heat is expended in heating the water, the

Table No. 41.—Efficiency of Fuels.

British Thermal Units in One Pound of Fuel.	Water Evaporated from and at 212° per Pound of Fuel. Theoretical.	Efficiency of Fuel. Theoretical.	EFFICIENCY OF BOILER.									
			90 PER CENT.		80 PER CENT.		70 PER CENT.		60 PER CENT.		50 PER CENT.	
			Water Evaporated from and at 212° per Pound of Fuel.	Efficiency of Fuel.	Water Evaporated from and at 212° per Pound of Fuel.	Efficiency of Fuel.	Water Evaporated from and at 212° per Pound of Fuel.	Efficiency of Fuel.	Water Evaporated from and at 212° per Pound of Fuel	Efficiency of Fuel.	Water Evaporated from and at 212° per Pound of Fuel.	Efficiency of Fuel.
14,650	15.2	100.	13.6	90.0	12.1	80.0	10.6	70.0	9.1	60.0	7.6	50.0
14,500	15.0	99.0	13.5	89.0	12.0	79.2	10.5	69.3	9.0	59.4	7.5	49.5
14,250	14.8	97.3	13.3	87.6	11.8	77.8	10.3	68.1	8.9	58.4	7.4	48.7
14,000	14.5	95.6	13.0	86.0	11.6	76.5	10.1	66.9	8.7	57.4	7.3	47.8
13,750	14.2	93.9	12.8	84.5	11.4	75.1	10.0	65.7	8.6	56.3	7 1	47.0
13,500	14.0	92.2	12.6	83.0	11.2	73.8	9.8	64.5	8.4	55.3	7.0	46.1
13.250	13.7	90.5	12.3	81.5	11.0	72.4	9.6	63.4	8.3	54.3	6.9	45.3
13,000	13.5	88.8	12.1	79.9	10.8	71.0	9.4	62.2	8.1	53.3	6.7	44.4
12,750	13.2	87.1	11.9	78.4	10.5	69.7	9.2	61.0	7.9	52.3	6.6	48.6
12,500	12.9	85.4	11.6	76.9	10.3	68.3	9.0	59.8	7.7	51.2	6.5	42.7
12,250	12.7	83.7	11.4	75.3	10.1	67.0	8.9	58.6	7.6	50.2	6.4	41.9
12,000	12.4	82.0	11.2	73.8	9.9	65.6	8.7	57.4	7.5	49.2	6.2	41.0
11,750	12.2	80.3	11.0	72.3	9.7	64.2	8.5	56.2	7.3	48.2	6.1	40.2
11,500	11.9	78.6	10.8	70.7	9.6	62.9	8.4	55.0	7.1	47.2	6.0	39.3
11,250	11.7	76 9	10.5	69.2	9.3	61.5	8.2	53.8	7.0	46.1	5.9	38.5
11,000	11.4	75.2	10.2	67.7	9.1	60.2	8.0	52.6	6.8	45.1	5.7	37.6
10,750	11.1	73.5	10.0	61.2	8.9	58.8	7.8	51.5	6.7	44.1	5.6	36.8
10,500	10.9	71.8	9.8	64.6	8.7	57.4	7.6	50.3	6.5	43.1	5.4	35.9
10,250	10.6	70.0	9.5	63.0	8.5	56.0	7.4	49.1	6.4	42.0	5.3	35.0
10,000	10.4	68.3	9.3	61.5	8.3	54.7	7.2	47.8	6.3	41.0	5.2	34.2
9,750	10.1	66.6	9.1	59.9	8.1	53.3	7.1	46.6	6.1	40.0	5.1	33.3
9,500	9.8	64.9	8.8	58.4	7.9	51.9	6.9	45.4	5.9	38.9	4.9	32.5
9,250	9.6	63.2	8.6	56.9	7.7	50.6	6.7	44.2	5.7	37.9	4.8	31.6
9,000	9.3	61.4	8.4	55.3	7.5	49.1	6.5	43.0	5.6	36.8	4.7	30.7
8,750	9.1	59.7	8.2	53.7	7.3	47.8	6.3	41.8	5.4	35.8	4.6	29.8
8,500	8.8	58.0	7.9	52.2	7.0	46.4	6.2	40.6	5.3	34.8	4.4	29.0

amount of heat required to evaporate one pound of water is equal to the latent heat of steam at atmospheric pressure; that is, 965.7 B. T. U. The evaporation per pound of coal or combustible, as reduced to this basis, is known as the *unit of evaporation.*

In illustration of the method of calculation, suppose that a given test indicates that 8.73 pounds of water fed to the boiler at 120° have been evaporated into steam of 83.3 pounds gauge pressure by the combustion of one pound of the fuel under test, without correction for moisture in steam and fuel. The absolute steam pressure is $83.3 + 14.7 = 98$ pounds, and per Table No. 6 the total amount of heat contained in one pound of steam of this pressure is 1,213.40 B. T. U.; while per Table No. 3 the total heat of the water of 120° temperature is 120.149 B. T. U. Evidently, then, the amount of heat which was imparted to one pound of water at 120° in order to convert it into steam of 83.3 pounds gauge pressure was $1{,}213.40 - 120.149 = 1{,}093.251$ B. T. U. As the latent heat of steam at atmospheric pressure is 965.7 B. T. U., the evaporation of 8.73 pounds of water under the stated conditions is equivalent to the evaporation of —

$$\frac{1{,}093.251}{965.7} \times 8.73 = 9.87 \text{ pounds,}$$

from and at 212°.

For ultimate comparison of fuels the results should be corrected for moisture in the steam and in the fuel. If, under the conditions of the test just used for illustration, the steam had contained 1.2 per cent and the coal 3.5 per cent of moisture, the method of correction would be as follows: —

The actual proportion of dry steam would be $100 - 1.2 = 98.8$ per cent, and that of dry coal, $100 - 3.5 = 96.5$ per cent. The amount of water evaporated into dry steam from and at 212° per pound of fuel would, therefore, be $9.87 \times 0.988 = 9.75$ pounds; and the evaporation of dry steam per pound of dry fuel would be $9.75 \div 0.965 = 10.10$ pounds; or, combined in one calculation,—

$$\frac{9.87 \times 0.988}{0.965} = 10.10 \text{ pounds.}$$

To ascertain the equivalent evaporation per pound of combustible in the fuel, the proportion of ash must be ascertained by careful weighing and deducted from the total fuel burned. If the ash in the fuel from which the preceding results were obtained had amounted to 6.4 per cent, the evaporation of water from and at 212° into dry steam would have been —

$$\frac{10.10}{1.00 - 0.064} = 10.79 \text{ pounds.}$$

Table No. 42, calculated by the method previously explained, presents a series of factors by means of any one of which, corresponding to the given temperature of feed water and pressure of steam, the evaporative result obtained may be transformed into the equivalent evaporation from and at 212°. Thus, taking the conditions of 83.3 pounds and 120° temperature already given, the factor (ascertained by interpolation) is 1.132, which, multiplied by 8.73, gives 9.88[1] pounds of water evaporated from and at 212°. Evidently, this table may be used in a converse manner to determine what conditions of feed temperature and steam pressure may be equivalent to a stated evaporation from and at 212°. Thus, for instance, an evaporation of 9.87 pounds from and at 212° is equivalent to $9.87 \div 1.149 = 8.59$ pounds from water at 100° into steam at 70 pounds gauge pressure.

Relative Efficiency of Various Coals. — Although the preceding applies to all classes of fuel, the greatest interest centres in the practical calorific value of various kinds of coal; for upon this fuel, above all others, is general steam-boiler practice most dependent. It must have already become evident that the apparent efficiency of the coal and of the boiler in connection with which it is burned are interdependent. Increased calorific value on the part of the coal insures an increase in the output of the boiler; while an improvement in the proportions of the boiler, its appurtenances or its method of operation, whereby its steaming power is increased per pound of coal, likewise raises the practical efficiency of the coal.

In general it may be stated that any furnace is well adapted to the combustion of anthracite and semi-bituminous coals containing less than 20 per cent of volatile matter. For coals containing between 20 and 40 per cent, a plain grate-bar furnace with firebrick arch thrown over it is desirable, because of its ability to keep the furnace chamber hot. For coals which contain over 40 per cent of volatile matter, a furnace is desirable which is surrounded by firebrick, with a large combustion chamber and special appliances for introducing very hot air to the gases distilled from the coal. A separate gas producer and combustion chamber, arranged for heating both air and gases before they unite, serves the same purpose.

The efficiency of a given coal is dependent, not only on its chemical composition and theoretical heat value, but to a great degree upon the percentage of ash and moisture which it contains, and upon the size of the respective pieces or particles, both absolutely and relatively to each other.

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[1] The slight difference between this and the calculated result is due to the fact that the numbers in the table are not carried out to more decimal places.

Table No. 42.—Factors of Evaporation.

Temperature of Feed Water, in Degrees Fahr.	Pressure in Pounds per Square Inch above the Atmosphere.											
	0	20	40	50	60	70	80	90	100	120	140	160
32°	1.187	1.201	1.211	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237
35	1.184	1.198	1.208	1.211	1.214	1.216	1.219	1.221	1.224	1.228	1.231	1.234
40	1.179	1.193	1.203	1.206	1.209	1.211	1.214	1.216	1.219	1.223	1.226	1.229
45	1.173	1.187	1.197	1.200	1.203	1.205	1.208	1.210	1.213	1.217	1.220	1.223
50	1.168	1.182	1.192	1.195	1.198	1.200	1.203	1.205	1.208	1.212	1.215	1.218
55	1.163	1.177	1.187	1.190	1.193	1.195	1.198	1.200	1.203	1.207	1.210	1.213
60	1.158	1.172	1.182	1.185	1.188	1.190	1.193	1.195	1.198	1.202	1.205	1.208
65	1.153	1.167	1.177	1.180	1.183	1.185	1.188	1.190	1.193	1.197	1.200	1.203
70	1.148	1.162	1.172	1.175	1.178	1.180	1.183	1.185	1.188	1.192	1.195	1.198
75	1.143	1.157	1.167	1.170	1.173	1.175	1.178	1.180	1.183	1.187	1.190	1.193
80	1.137	1.151	1.161	1.164	1.167	1.169	1.172	1.174	1.177	1.181	1.184	1.187
85	1.132	1.146	1.156	1.159	1.162	1.164	1.167	1.169	1.172	1.176	1.179	1.182
90	1.127	1.141	1.151	1.154	1.157	1.159	1.162	1.164	1.167	1.171	1.174	1.177
95	1.122	1.136	1.146	1.149	1.152	1.154	1.157	1.159	1.162	1.166	1.169	1.172
100	1.117	1.131	1.141	1.144	1.147	1.149	1.152	1.154	1.157	1.161	1.164	1.167
105	1.111	1.125	1.135	1.139	1.141	1.143	1.146	1.148	1.151	1.155	1.158	1.161
110	1.106	1.120	1.130	1.133	1.136	1.138	1.141	1.143	1.146	1.150	1.153	1.156
115	1.101	1.115	1.125	1.128	1.131	1.133	1.136	1.138	1.141	1.145	1.148	1.151
120	1.096	1.110	1.120	1.123	1.126	1.128	1.131	1.133	1.136	1.140	1.143	1.146
125	1.091	1.105	1.115	1.118	1.121	1.123	1.126	1.128	1.131	1.135	1.138	1.141
130	1.085	1.099	1.109	1.112	1.115	1.117	1.120	1.122	1.125	1.129	1.132	1.135
135	1.080	1.094	1.104	1.107	1.110	1.112	1.115	1.117	1.120	1.124	1.127	1.130
140	1.075	1.089	1.099	1.102	1.105	1.107	1.110	1.112	1.115	1.119	1.122	1.125
145	1.070	1.084	1.094	1.097	1.100	1.102	1.105	1.107	1.110	1.114	1.117	1.120
150	1.065	1.079	1.089	1.092	1.095	1.097	1.100	1.102	1.105	1.109	1.112	1.115
155	1.059	1.073	1.083	1.086	1.089	1.091	1.094	1.096	1.099	1.103	1.106	1.109
160	1.054	1.068	1.078	1.081	1.084	1.086	1.089	1.091	1.094	1.098	1.101	1.104
165	1.049	1.063	1.073	1.076	1.079	1.081	1.084	1.086	1.089	1.093	1.096	1.099
170	1.044	1.058	1.068	1.071	1.074	1.076	1.079	1.081	1.084	1.088	1.091	1.094
175	1.039	1.053	1.063	1.066	1.069	1.071	1.074	1.076	1.079	1.083	1.086	1.089
180	1.033	1.047	1.057	1.060	1.063	1.065	1.068	1.070	1.073	1.077	1.080	1.083
185	1.028	1.042	1.052	1.055	1.058	1.060	1.063	1.065	1.068	1.072	1.075	10.78
190	1.023	1.037	1.047	1.050	1.053	1.055	1.058	1.060	1.063	1.067	1.070	1.073
195	1.018	1.032	1.042	1.045	1.048	1.050	1.053	1.055	1.058	1.062	1.065	1.068
200	1.013	1.027	1.037	1.040	1.043	1.045	1.048	1.050	1.053	1.057	1.060	1.063
205	1.008	1.022	1.032	1.035	1.038	1.040	1.043	1.045	1.048	1.052	1.055	1.058
210	1.004	1.017	1.027	1.030	1.033	1.035	1.038	1.040	1.043	1.047	1.050	1.053
212	1.002	1.000	—	—	—	—	—	—	—	—	—	—

An efficiency of 100 per cent on the part of either the coal or the boiler is an absolute impossibility because of certain losses which are incident to the combustion of the coal and the operation of the boiler. Of these losses some are inevitable, while others may be diminished or avoided.

The unavoidable losses are: —

First. The heat loss by converting into steam the water contained in the coal, and in the air used in burning it, as well as that formed by the burning of the hydrogen and the heating of the steam thus formed to the temperature at which the gases leave the stack.

Second. The heat necessary to raise to the stack temperature the carbonic acid gas formed by burning the carbon, the nitrogen originally present in the air from which the oxygen has been taken to form carbonic acid, the sulphurous acid and the excess of air which is supplied to secure perfect combustion. There is also a loss through heat in the hot ashes removed from the ash pit, as well as through the unconsumed carbon remaining in the ash. A further loss occurs through radiation from the boilers and walls, which can, by careful construction and covering, be greatly reduced but never eliminated.

The losses which are more or less avoidable are: —

First. Those due to incomplete combustion, as evidenced in the presence of smoke and carbonic oxide in the flue gases and in unconsumed coal in the ashes. This latter loss is due to the original small size or the subsequent decrepitation of the coal, which results in the dropping of more or less of it through the grates without being consumed. In addition a small amount of hydrogen or marsh gas may pass out with the gases.

Second. Loss from excess of air, due to the fact that to secure practically perfect combustion air must be supplied considerably in excess of the theoretical quantity chemically required for combustion. This loss is twofold, being dependent upon the quantity of unused oxygen and associated nitrogen and upon the moisture in the air.

Third. The loss resulting from too high temperature of the gases leaving the boiler. This loss, except in so far as it is influenced by the air supply and the rate of combustion, is dependent upon the design of the boiler and its appurtenances, and, therefore, is not chargeable to the character of the fuel. It is one of the most important factors in fuel efficiency.

Fourth. Loss of heat by removing ashes at too high a temperature. This, by care, may be reduced but not entirely avoided.

Fifth. Loss by radiation. This may be reduced by increasing the thickness of walls and covering all exposed portions of the boiler. But from a practical standpoint it can never be entirely avoided.

The influence of these various sources of loss upon the efficiency of fuels and boilers will be considered in succeeding pages. Independent of such consideration the relative efficiency of various coals, as indicated by comparative tests, may, however, be here introduced. It is already evident that, for the purposes of strict comparison of evaporative powers, coals should be tested under identical conditions. What is more, all ordinary grades of coal should be tested under such a variety of boilers and rates of combustion, air supply and draft that they may be intelligently compared one with the other under all ordinary conditions. Such extensive and strictly comparable results do not, as yet, exist, although already much careful work has been and is now being done to furnish such a basis of comparison.

At the present time it is only possible to compare with each other the results in certain groups of tests, but only to a limited extent to correspondingly compare the results in one group with those in another. By such comparisons as are allowable, it is possible to approximate with reasonable accuracy to the relative values of the coals under consideration.

A series of such results, compiled from the reports of numerous tests by Mr. George H. Barrus,[1] is presented in Table No. 43. In harmony with the results obtained in a large number of tests, the evaporation per pound of combustible in anthracite broken coal has been taken, as a standard of comparison, in round numbers at 11.0 pounds from and at 212°. With an average of 11 per cent of ash this is equivalent to 9.79 pounds of water evaporated from and at 212° by one pound of coal.

Table No. 43.—Comparative Evaporative Efficiency of Various Coals.

KIND OF COAL.	Water Evaporated from and at 212° by One Pound of Dry Coal.	Relative Efficiency in per cent. Cumberland = 100.
Cumberland,	11.04	100
Anthracite, Broken,	9.79	89
Anthracite, Chestnut,	9.40	85
Two parts Pea and Dust and one part Cumberland,	9.38	85
Two parts Pea and Dust and one part Culm,	9.01	82
Anthracite, Pea,	8.86	80
Nova Scotia Culm,	8.42	76

Other comparisons of efficiency between different kinds of fuel may be drawn from the various reports of tests upon succeeding pages.

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[1] Boiler Tests. George H. Barrus. Boston, 1891.

Influence of Ash. — Beyond its indication of the relative presence of incombustible matter, the influence of ash upon the thermal efficiency of coal is threefold. Its presence is the measure of the loss through the amount of unconsumed carbon which it contains, through the heat lost when the ash is removed in a heated condition, and through the influence of the ash in clogging the fire and preventing free combustion. From a commercial standpoint its presence, furthermore, proportionately increases the original cost of freight and handling per heat unit derived, as well as the subsequent expense incident to its own removal and transportation to a proper dumping-place, — a fact which demands careful consideration.

The percentage of carbon, either in the form of cinder or decrepitated coal, which eventually forms a part of the ash, can be somewhat reduced by skilful manipulation of the fire. But as the greatest saving of such carbon lies in slower and gentler firing and in admitting more air, the efficiency, as a whole, is liable to be lowered rather than raised if the attempt to economize is carried to an extreme. The carbon ordinarily present in ash varies greatly, but may be broadly stated to range between 10 and 60 per cent. The loss by carbon in the ash is, therefore, dependent upon the percentage of ash in the coal, as is shown in Table No. 44.

Table No. 44. — Loss of Fixed Carbon on Account of Carbon in Ash.

Per cent of Carbon in Ash.	Per Cent of Ash in Coal Fired.									
	9 per cent.	10 per cent.	11 per cent.	12 per cent.	13 per cent.	14 per cent.	15 per cent.	16 per cent.	17 per cent.	18 per cent.
75	31.76	35.70	39.73	43.88	48.14	52.47	56.94	61.51	66.23	71.05
70	24.70	27.75	30.89	34.11	37.43	40.80	44.27	47.83	51.51	55.26
65	19.65	22.09	24.59	27.16	29.79	32.48	35.23	37.07	41.00	43.98
60	15.87	17.84	19.86	21.93	24.07	26.23	28.47	30.76	33.11	35.52
55	12.94	14.54	16.18	17.87	19.60	21.37	23.19	24.05	26.98	28.94
50	10.59	11.90	13.24	14.63	16.04	17.49	18.98	20.50	22.08	23.68
45	8.66	9.76	10.83	11.97	13.12	14.31	15.52	16.77	18.06	19.38
40	7.05	7.92	8.82	9.74	10.69	11.66	12.64	13.66	14.72	15.79
35	5.69	6.40	7.12	7.87	8.63	9.40	10.21	11.04	11.88	12.75
30	4.53	5.10	5.66	6.27	6.87	7.43	8.12	8.78	9.45	10.14
25	3.52	3.96	4.41	4.96	5.34	5.82	6.31	6.83	7.35	7.89
20	2.64	2.97	3.30	3.65	4.02	4.36	4.74	5.13	5.52	5.92
15	1.86	2.09	2.33	2.57	2.82	3.09	3.35	3.61	3.89	4.18
10	1.17	1.32	1.47	1.62	1.77	1.93	2.10	2.27	2.45	2.62
5	0.56	0.63	0.69	0.76	0.84	0.91	0.99	1.07	1.15	1.24
1	0.10	0.12	0.13	0.14	0.16	0.17	0.18	0.20	0.22	0.24

It is obvious that the greater the amount of hot ashes resulting from the combustion of a given amount of coal, the greater will be the loss of heat when these ashes are removed. As a means of clogging the grates, preventing free combustion and necessitating extra work on the part of the fireman, with a resulting excess of air while the fire doors are open, a large amount of ash in the fuel exerts a very important influence.

Influence of Moisture in Coal. — Moisture in coal is an exceedingly variable quantity, depending upon the character of the coal, its temperature and its previous exposure to the atmosphere. Under ordinary conditions its percentage varies from 1 to 5 per cent. Whatever its amount, it must all be raised to 212°, evaporated into steam and the steam raised to the temperature of the escaping gases. It, therefore, has an important influence upon the theoretical heat value of a given coal. Thus, if one coal was composed of 80 per cent carbon, 15 per cent ash and 5 per cent water, and another consisted of the same proportion of carbon, with 5 per cent ash and 15 per cent water, the theoretical calorific value — viz., 11,720 B. T. U.— would be the same, being directly dependent upon the amount of carbon. But in the first case the available heat (neglecting losses not due to water) would be 10,600 B. T. U., while in the second it would be 10,488 B. T. U., if the waste gases were assumed to escape at 500°.

Influence of Size of Coal. — A still further, and under certain conditions a very important, influence upon the efficiency of coals, particularly the anthracites, is exerted by the size of their respective pieces or particles. For the freest burning they should be as nearly of a size as possible; hence, the screening process at the mines and their sale in stated sizes. In the smaller sizes of anthracite, consisting of culm and screenings, or slack, the inherent dust and minute particles render them difficult coals to burn unless mixed with a certain proportion of bituminous coal and burned upon special grates, with an intensity of draft which can only be economically produced by mechanical means. This feature becomes more pronounced as the coal becomes finer, and makes legitimate comparison with other coals a somewhat difficult matter.

Broadly stated, the requisites to success in the combustion of small anthracites are, as expressed and explained by Mr. Geo. H. Ward,[1] "first, draft; and, second, manipulation of the fires. Of course no fire will burn without draft, and the greater the amount of fire in a given space the stronger the draft must be to properly consume the fuel. With the larger sizes of fuel this question of draft is less prominent, but when we come to burn the smaller sizes the draft is of the utmost importance. The coal will pack on the grate, and, owing to the way the pieces will arrange themselves one to another, it will be impossible to get sufficient air through the bed of fire unless the draft is strong enough to displace the smaller particles. . . . Another and perhaps a better reason is that the proportion of ash is somewhat greater with the smaller coals, and as the fire is but a thin crust on top of this, it follows that a somewhat stronger draft will be required to get the necessary volume of air through the bed of ashes and the closely packed crust of the fire on top."

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[1] The Economy of Small Anthracite Coals. Geo. H. Ward. A paper read before the Engineers' Club of Brooklyn, N. Y. 1892.

The matter of burning the smaller sizes of refuse coal will be still further considered from a commercial standpoint, but the conditions which control the successful combustion of such fuel may be here discussed. While the terms "screenings" and "slack" are generally applied to the refuse of local coal yards, the term "culm" is restricted in its application to the refuse or waste from the anthracite coal mines. Originally applied to any mixture below pea coal, this term has become more restricted in its meaning, as the smaller sizes have been removed in the ordinary process of screening, until at the present time it is in some localities applied to that which cannot be sold as buckwheat, but which has had a part of the dust washed out. It is, therefore, evident that in any discussion of the use of a refuse fuel more than its name is necessary to determine its exact character.

Although there is considerable variety in the size classification of coal at different mines, the generally accepted dimensions, as determined by the limiting diameters of the perforations in the screens, are about as presented in Table No. 45.

Table No. 45. — Sizes of Coal.

Designation.	Diameter of Perforation over which Coal will pass.	Diameter of Perforation through which Coal will pass.
Dust		3-32 inch.
No. 3 Buckwheat	3-32 inch.	3-16 "
Bird's-eye	1-8 "	5-16 "
No. 2 Buckwheat, or Rice	3-16 "	3-8 "
No. 1 Buckwheat	3-8 "	9-16 "
Pea	9-16 "	7-8 "
Chestnut	5-8 "	1⅛ "
Small Stove	1 "	1¾ "
Large Stove	1¾ "	2¼ "
Egg	2¼ "	2¾ "
Broken	2¾ "	4 "
Steamboat	4 "	7 "

The important factors in the successful burning of small anthracite coals are: —

First. Mechanical draft, preferably applied beneath the grate.

Second. Large grate area.

Third. Grate constructed with a practically plain surface to prevent lodgement of coal.

Fourth. Grate of proper design for ready removal of ash and clinker.

Fifth. Air spaces not over $\frac{1}{16}$ to $\frac{3}{16}$ inch wide, except for No. 1 buckwheat or bituminous slack, for which they may be $\frac{1}{4}$ inch wide.

Sixth. Arrangements to allow of the feeding and cleaning of fires without excessive opening of doors or dropping of ashes into ashpit.

Seventh. Thin fires and frequent and careful firing. The thickness of the bed should diminish with the rate of combustion.

Eighth. Reduction of draft above the fire as the rate of combustion decreases. The fine character of the material, as a result of which its particles pack closely together with but small interstitial spaces, makes strong draft imperative in order to secure the passage of the proper amount of air through the bed of fuel. That mechanical draft is the necessary and most economical means of securing this result is the verdict of all who have had experience in the burning of such fuel. Owing to the usually somewhat complicated grate or feeding arrangements for the burning of small anthracites, forced draft is usually applied beneath the grates, but may under certain conditions be favorably assisted by supplementary induced draft. The careful and extensive experiments of Mr. Eckley B. Coxe, upon the burning of small anthracite coals, have served to throw considerable light upon the features and requirements of such undertakings. He concludes[1] "that it is possible that the best results in burning these small coals may be obtained by using a blower under the grate and a suction apparatus in the stack."

A large grate area permits of the carrying of thinner fires with a given total consumption of coal, while the small spaces through the grate are necessary to prevent too great loss by the dropping of fine coal through them. The necessity of such grates is shown in the report of the commission[2] of which Mr. Coxe was a member and for whom these experiments were instituted: "A careful study of the burning of culm, — that is, the burning of coals with more or less dust in them, — in these and other experiments, seemed to show that in almost all cases it is accompanied by a very high percentage of carbon in the ash, which analysis showed, in some cases, reached 58 per cent. Unless special precautions are taken to prevent it, a large portion of the fine coal runs down through the grate. When culm gets red hot it acts almost like dry sand and works its way into the ashpit, thus increasing largely the percentage of carbon. When coal has to be transported any distance, the value of the culm at the mines being very small, it is probable, from the investigations made, that it would be cheaper to remove the dust and transport only the larger coal."

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[1] Furnace for Burning Small Anthracite Coals. Eckley B. Coxe. Trans. Am. Inst. Mining Engineers, Vol. XXII. 1894.

[2] Commonwealth of Pennsylvania. Report of Commission Appointed to Investigate the Waste of Coal Mining, with the View to the Utilizing the Waste. May, 1893.

It appears that the combustion of small anthracites is more perfect when the coal remains undisturbed, or as nearly as possible in the condition in which it was put upon the fire, instead of being turned over so that the partially consumed and unconsumed coal are mixed together. For such fuel the travelling grate is particularly adapted, as it leaves the fuel undisturbed, and makes possible a gradation of the draft to meet the varying conditions incident to the progress of the grate and the combustion of the fuel upon it.

Whatever the size of these smaller grades of fuel, certain special furnace arrangements are necessary. The smaller they are the more extensive and expensive the appliances; and for each special arrangements are necessary. It is, therefore, undesirable and positively uneconomical to mix the sizes. Thus a mixture of dust with pea or chestnut coal, while burning more freely because of the easier passage of air through it, will not give so good evaporative results as an intermediate size containing less of the larger and the smaller pieces or dust; that is, having all of its pieces nearer a size. This difference is due to the fact that in the mixture the fine coal is completely consumed before the larger pieces.

The advantage of mixing a slight amount of bituminous coal with the smaller anthracites is well known. The somewhat glutinous character of the former, when burning, serves to make a more coherent mass of the entire body of fuel, thereby preventing the dust from being blown through the flues or dropped through the grates, while keeping the bed more open and thereby increasing the rate of combustion. Culm and rice coal thus fired give results in total evaporation which cannot be reached by the same anthracites alone.

The experiments of Mr. Coxe have demonstrated that "the temperature developed by the burning of the smaller coals decreases with the size of the coal; this naturally involves a larger heating surface in the boiler in order to develop the same number of horse-powers; that is to say, if you are burning pea coal, and obtaining one horse-power for every nine square feet of heating surface, you would probably require from 20 to 25 per cent more heating surface if you are using No. 3 buckwheat; although you may be evaporating practically the same amount of water per pound of coal."

Table No. 46 gives results of tests[1] of small anthracites by Mr. Coxe.

Table No. 46.—Results of Tests of Pea and Buckwheat Coals.

ITEMS.		No. of Test and Kind of Fuel.				
		1 Oneida Pea Coal.	2 Oneida No. 1 Buckwh't.	3 Oneida No. 2 Buckwh't.	4 Oneida No. 3 Buckwh't.	5 Eckley No. 3 Buckwh't.
Pounds of water evaporated per pound of dry coal, actual conditions,		7.14	6.62	7.17	7.21	7.36
Pounds of water evaporated per pound of dry coal from and at 212°,		8.56	7.94	8.60	8.65	8.74
Pounds of water evaporated per pound of combustible from and at 212°,		10.14	10.06	10.57	11.12	11.10
Pounds of water evaporated from and at 212° per square foot of heating surface,		3.70	3.21	3.13	3.13	3.06
Pounds of coal per square foot of grate per hour,		13.63	13.58	11.40	11.34	9.44
Average temperature of escaping gases,		549°	543°	498°	503°	372°
Moisture in steam, per cent,		2.2	2.0	1.9	1.9	——
Moisture in coal as fired, per cent,		2.63	4.06	8.62	6.53	4.93
Per cent of ash,		15.60	20.10	18.71	22.27	21.3
Per cent of carbon in ash,		15.85	12.35	9.33	31.90	29.63
Average pressure of blast in inches of water in entrance chamber,		0.375	0.5	0.625	1.04	1.125
Analysis of coal.	Water at 225°,	2.15	2.00	2.10	2.05	2.50
	Volatile combustible matter,	5.10	4.90	5.45	5.42	5.00
	Ash,	12.55	17.35	15.50	12.90	13.97
	Carbon, fixed,	80.20	75.75	76.95	79.63	78.53
	Specific gravity,	1.620	1.664	1.655	1.642	1.665
Sizing test.	Chestnut, over 7/8 inch round mesh,	8.44	0.98	——	——	——
	Pea coal, between 7/8 inch and 9-16 inch round mesh,	60.65	6.85	0.31	1.50	1.21
	No. 1 Buckwheat, between 9-16 inch and 3/8 inch round mesh,	21.70	57.72	4.76	4.58	2.60
	No. 2 Buckwheat, between 3/8 inch and 3-16 inch round mesh,	3.68	28.74	66.57	17.75	31.94
	No. 3 Buckwheat, between 3-16 inch and 3-32 inch round mesh,	1.40	2.39	19.87	45.95	49.57
	Between 3-32 inch and 1-16 inch, sometimes allowed in No. 3 Buckwheat,	4.13	1.49	2.39	19.79	6.31
	Dust through 1-16 inch round mesh,	——	1.83	6.10	10.43	8.37
Slate test,	Pure coal, specific gravity below 1.70,	92.00	76.18	78.28	86.98	83.85
	Slate and bone, specific gravity above 1.70,	8.00	23.82	21.72	13.02	16.15

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[1] Furnace for Burning Small Anthracite Coals. Eckley B. Coxe. Am. Inst. Mining Engineers, Vol. XXII. 1894.

Tests 1 to 4 inclusive were made upon two Stirling water-tube boilers, each having 1,725 square feet of heating surface and 55 square feet of grate; while No. 5 was made upon a cylinder boiler with drums and connecting tubes, the total heating surface being 1,862 square feet and 68.75 square feet of grate. All the boilers were equipped with Coxe travelling grates and forced draft under the grate.

Other tests,[1] upon a return tubular boiler equipped with Coxe stoker, indicated the relative efficiencies of various fuels as presented in Table No. 47.

Table No. 47. — Relative Efficiencies of Small Anthracite Coals.

Kind of Coal.	Pounds of Water per Pound of Coal from and at 212°.	Pounds of Water per Pound of Combustible from and at 212°.
Buckwheat,	8.77	11.07
Rice (No. 2 Buckwheat),	9.05	11.18
Culm (Pea, Buckwheat, Rice, Barley, Dust),	8.74	11.19
Barley (No. 3 Buckwheat),	8.39	10.89

Influence of Air Supply. — Although the loss or waste that occurs through an improper amount of air supplied for combustion is not properly chargeable to the fuel, yet, under practical conditions, this factor exerts an important influence upon their relative efficiency. As, with all other conditions the same, the amount of air passing through the fire is with chimney draft liable to change according to the atmospheric conditions and the methods of firing and operating the dampers, especial care is necessary to secure uniformity and equality of air volume in all tests which are to be compared. Such uniformity can be readily maintained in plants operated by mechanical draft.

In reality the intensity of the draft whereby the requisite volume of air is supplied is of the greatest importance, as is clearly shown in succeeding chapters. This makes possible thick fires and the most economical distribution of the fuel. "A considerable saving of fuel," as asserted by Hutton,[2] "may be effected by the employment of well-arranged forced draft, . . . and greater power may be obtained with less size and number of boilers than with boilers having combustion with natural draft."

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[1] Some Thoughts on the Economical Production of Steam, with Special Reference to the Use of Cheap Fuel, by a Miner of Coal. Eckley B. Coxe. Trans. New England Cotton Manufacturers' Association. April, 1895.

[2] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

The method of calculating the theoretical amount of air required for the complete combustion of any fuel has already been explained. If less than this amount is supplied, a certain portion of the carbon passes off unconsumed and forms smoke; while a part of the remainder, being insufficiently supplied with oxygen, forms carbonic oxide, the product of incomplete combustion. If the air be supplied in excess of that necessary for perfect combustion, there is a definite loss, which is twofold in its character: First, the excess of air entering the furnace is heated by the burning fuel, thereby lowering the temperature of the mixture of gases and air below that which would prevail if the gases only were present. As a consequence, the rate of absorption of heat by the water is reduced, for it is dependent upon the difference in temperature between the water and the gases. Second, owing to larger volume and higher velocity the temperature of the mixture of gases and air escaping to the chimney is higher than would be the case if there were no excess of air; while the increased volume is such that the total amount of heat thus carried away, without exerting any useful effect, is greatly increased. In other words, paradoxical as it may seem, the larger the volume of air supplied, the higher will be the temperature of the escaping gases.

Influence of the Frequency of Firing. — While the rate at which any given coal should be fed to the furnace is largely dependent upon the character of the coal, nevertheless it is doubtless true that in most cases it is fed in too large quantities and at too long intervals. The natural result is a series of decided and almost critical changes in the condition of the fire, to be compared to the effect of eating a large amount of food once a day instead of a smaller amount at a greater number of times. It is evident that efficiency, as regards the frequency of firing, is entirely dependent upon the fireman, and hence for favorable conditions the advantages of a mechanical stoker.

For the purpose of ascertaining, so far as possible, the relative results of different rates of firing, M. Burnat[1] conducted a series of experiments extending over eight weeks, with the same fireman and the same boiler.

The general results are presented in Table No. 48. The advantage of the smallest charge of 13 pounds over the maximum of 55 pounds is in the first series 3.03 per cent, and in the second 8.19 per cent. These results are obtained, notwithstanding the fact that with the more frequent firing the doors were more frequently opened. A boiler arranged so that the damper became closed or nearly so when the door was opened showed upon test an increased evaporation of 14 to 15 per cent due to this arrangement.

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[1] Bulletin de la Société Industrielle de Mulhouse, Vol. XLVI. 1876.

Table No. 48. — Influence of Frequency of the Charges of Coal.

KIND OF COAL.	Cubic Feet of Air at 62° per Pound of Coal.	Temperature of Feed Water.		Temperature of Hot Gases.		Pounds of Coal Consumed.			Per Cent of Residue	Pounds of Water Evaporated from and at 212° per Pound of Coal.
		Entering Feed Heater.	Entering Boiler.	Leaving Boiler.	Leaving Feed Heater.	Per Hour.	Per Square Foot of Grate.	Per Charge		
Ronchamp, Nut,	202	90°	213°	849°	421°	225	10.8	13.3	12.9	9.87
	202	87	224	840	426	225	10.8	26.6	13.4	9.59
	197	87.5	227	844	414	225	10.8	39.2	12.8	9.59
	202	86	226	835	421	225	10.8	55.4	12.8	9.58
Ronchamp, large and small,	226	87°	230°	779°	396°	225	10.8	55.0	16.1	8.91
	212	87	226	784	410	225	10.8	41.1	14.6	9.18
	201	87.5	229	795	410	225	10.8	28.0	14.7	9.38
	202	86	228	853	489	225	10.8	15.0	12.6	9.64

Loss on Account of Smoke. — The loss resulting from the formation of smoke is absolute; for it is equivalent to directly robbing the fire of a part of the fuel from which not only has no heating effect been secured, but upon which heat has actually been wasted in raising it to the temperature of the escaping flue gases. Notwithstanding the prevailing impression as to the great losses due to the formation of smoke, the actual waste is comparatively insignificant, as is shown by the following results of carefully conducted experiments by Mr. J. C. Hoadley.[1]

During an entire week gas was drawn from the flue of the boiler under test, passed through a gas meter and thence through a muslin strainer at the bottom of a vessel of water. When a sufficient quantity of the gases had been passed the water was evaporated, and the residuum was dried and weighed. The coal used was bituminous, of the following average composition: —

Carbon	81.03	per cent.
Hydrogen	3.84	"
Ash	7.19	"
Water	0.63	"
Oxygen	4.49	"
Nitrogen	2.00	"
Sulphur	0.82	"

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

The total quantity of coal burned during the week was 12,890 pounds, the total quantity of flue gases reduced to 72° being 4,263,119 cubic feet, and the total amount of solid matter 42.63 pounds, as shown by the test. There was, therefore, present in solid form in the flue gases only —

$$\frac{42.63}{12,890} = .0033 = 0.33 \text{ per cent}$$

of the matter originally present in the coal. As the gray color of the substance thus recovered indicated that it was not more than half carbon, it is evident that under the conditions of the test the proportion of carbon which was actually carried off in black smoke was about one-sixth of one per cent of the original coal.

MM. Scheurer-Kestner and Meunier[1] passed flue gases through asbestos, upon which the particles of carbon were deposited, and found that, with good fire and draft and an air supply of 257 cubic feet per pound of coal, one half of one per cent of the carbon of the coal was lost as smoke particles. In a second trial, with poor fire and draft and only 118 cubic feet of air per pound of coal, the deposited carbon was one per cent of the total contained in the coal. This latter is to be taken as a maximum, the conditions being decidedly adverse, and is more than would be produced in ordinary practice. The inference from these experiments must be that the average loss of carbon in the solid form as smoke may be taken not to exceed one half to three quarters of one per cent.

Loss on Account of Carbonic Oxide. — The loss of efficiency which ensues from the escape of carbonic oxide unconverted into carbonic acid is due to the much smaller amount of heat given out upon the incomplete combustion of carbon into carbonic oxide. While carbon burned to carbonic acid generates 14,650 B. T. U., the same quantity burned to carbonic oxide gives out only 4,400 B. T. U. For every pound of carbon which passes off in the form of carbonic oxide, there is, therefore, a loss of 14,650 — 4,400 = 10,150 B. T. U., or —

$$\frac{10,150}{14,650} = 0.6928 = 69.28 \text{ per cent.}$$

The ultimate effect of the formation of carbonic oxide upon the total heat of combustion can be best illustrated by reference to the calculation under "Heat of Combustion," in Chapter III. If, instead of the entire 0.80 pound of carbon having been perfectly burned, only 0.70 pound had entered into combustion with oxygen to form carbonic acid, and the remaining 0.10 pound had entered into combustion as carbonic oxide, the calculation would have been as follows:

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[1] Bulletin de la Société Industrielle de Mulhouse. 1868, 1869.

$$\text{Heat, in B. T. U.,} = 14{,}650 \times 0.70 + 4{,}400 \times 0.10 + 62{,}100\,(0.05 - \frac{0.027}{0.8})$$
$$= 13{,}590 \text{ B. T. U.;}$$

showing that there would be a loss in the calorific value of the coal of about 7 per cent. Had only half of the carbon been completely burned, the loss would have been about 18 per cent. But such a loss even as that first instanced does not usually occur continuously in any well designed and operated boiler furnace. In fact, the loss from this source appears to be largely overestimated in most cases.

Three days' continuous analysis of the flue gases from the boiler plant of the B. F. Sturtevant Company at Jamaica Plain, Mass., operating under induced draft, failed to indicate the presence of an amount of carbonic oxide greater than one tenth of one per cent, or sufficiently large to be identified by the refined apparatus employed for the purpose. In Hoadley's warm-blast furnace tests the carbonic oxide "never in the day time exceeded half of one per cent, and rarely exceeded half that small quantity when the dampers were open, for six weeks together."

From the analyses of the chimney gases of 124 boiler tests made at the Industrial Exhibition at Düsseldorf in 1880,[1] the average amount of carbonic oxide by volume was found to be 0.747 per cent. In all cases but one bituminous coal was used. The air supply was almost constant; the minimum amount of nitrogen shown in any of the tests being 79.3 per cent, and the maximum 84.08 per cent, but ranging in most cases between 80 and 81 per cent. Omitting four analyses in which the carbonic oxide ranged from 3 to 5.3 per cent, the average presence of this gas in the remaining 120 tests appears to have been 0.637 per cent.

There are some conditions of boiler practice, however, in which such good results do not obtain, and in which more or less serious losses may occur, largely due to an insufficiency of air. It should be clearly understood, however, that the same amount of air is not always required. When the coal upon the grate is thoroughly ignited the minimum supply is necessary, but when the fire is suddenly thickened and cooled by additional coal there is a demand for additional supply for the purposes of combustion, together with a tendency to clog the passages through which the air has previously passed, and thereby to prevent complete combustion at the surface of the fire. At this time, for perfect

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[1] Die Untersuchungen an Dampfmaschinen und Dampfkesseln, und an einigen Rheinischen und Westfälischen Kohlensorten auf der Gewerbe-Ausstellung in Düsseldorf in 1880, herausgegeben von H. v. Reiche und F. Böcking, Aachen, 1881. Velag von J. A. Meyer.

Table No. 49.—Carbonic Oxide Produced by Excessive Firing.

TIME.	Pounds of Coal Thrown on Grate.	Carbonic Acid in Chimney Gases.	Carbonic Oxide in Chimney Gases.	Ratio of Carbon in Carbonic Oxide to Total Carbon.	Pounds of Air per Pound of Coal.	Pounds of Coal Burned Each Half-Hour.	Ratio of Loss by Carbonic Oxide to Full Power of Coal.
H. M.	Pounds.	Per cent CO_2.	Per cent CO.	Per cent.	Pounds.	Pounds.	Per cent.
6:15 a. m.	200						
6:45	200						
7:15	200						
7:45	200						
8:15	200						
8:45	200						
9:00		5.12	2.54	43.80	33.2	83.81	27.84
9:15	200						
9:30		5.55	2.99	45.85	29.5	93.75	29.14
9:45	200						
10:00		7.79	3.99	44.63	21.4	129.24	28.37
10:15	200						
10:30		7.70	4.61	48.47	20.1	137.60	30.81
10:45	200						
11:00		7.82	4.70	48.57	19.8	139.68	30.88
11:15	200						
11:30		8.01	4.81	48.55	19.3	143.30	30.86
12:00 m.					19.3	143.30	
12:30 p. m.		15.21	0.25	2.52	19.3	143.30	1.60
12:45	200						
1:00					20.05	137.94	
1:30		14.11	0.21	2.28	20.8	132.96	1.49
2:00					21.05	137.94	
2:30		13.62	0.33	3.67	21.3	129.85	2.31
2:45	200						
3:00		14.50	0.48	4.95	19.4	142.56	3.14
3:30		13.18	0.29	3.34	22.0	125.34	2.12
3:45	200						
4:00		14.96	0.38	3.84	19.3	143.30	2.44
4:30		14.18	0.41	4.35	20.3	136.25	2.76
4:45	200						
5:00		13.01	0.41	4.72	22.0	125.34	3.00
Mean quantity of air					21.653		
Mean of all but first two					20.36		
Mean ratio of loss; first six, per cent							29.65
Mean ratio of loss; last eight, per cent							2.36

conditions, more air should be admitted, as is possible under the positive action of mechanical draft arranged to operate automatically. But such additional supply is only necessary for a comparatively short time.

The effect of excessive firing is practically equivalent to the reduction of draft and air supply for the regular amount of coal, and is conducive to the production of carbonic oxide. This was well exemplified by Mr. J. C. Hoadley in a special test,[1] the results of which are presented in Table No. 49. As is evident from the items in the first and second columns, the firing was at first very rapid, decreasing as the day passed. During the morning 200 pounds were fired every half-hour, while during the afternoon the same amount was fired hourly. Gas samples were not taken until 9 A. M., after which time the excess of carbonic oxide is noticeable until the slower firing began, when the carbonic oxide is immediately reduced, while the carbonic acid is practically doubled. The ratio of loss by carbonic oxide, indicated in the last column, is of special interest.

Admission of Air above the Fire. — Since the days of C. Wye Williams and his famous work on the "Combustion of Coal and the Prevention of Smoke," the introduction of air above or beyond the fire has been one of the favorite methods adopted in the attempt to perfect the combustion of coal and prevent the appearance of smoke. In so far as the admission of air for the purpose of perfecting the combustion is concerned, the results in the way of water evaporation per pound of fuel should naturally be looked to as the true indication of its efficiency. That, under certain conditions, increased efficiency can be thus obtained is evidenced in many tests. As a rule the gain is not large, but is greatest with bituminous coals, whose large percentage of volatile constituents tends more readily to the formation of both carbonic oxide and smoke.

Table No. 50, compiled from carefully conducted tests by Mr. George H. Barrus,[2] serves to show both the percentage of gain and loss by such contrivances, the character of which is indicated under the designating letters, as follows: —

A. Air conducted direct from outside the setting to the interior of the bridge wall and discharged therefrom through perforations in its top covering. The air thus supplied mingles with the lower strata of burning gas as it skims over the bridge.

B. Air supplied first to a pipe laid in the bridge wall and thence to perforated cast-iron globes resting upon its top. A more thorough mixture of air and gases is thus secured. A jet of steam is employed to increase the volume of air which would be drawn in by natural means, and the steam thus supplied mingles with the air.

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[1] Warm-Blast Steam-Boiler Furnace. New York, 1886.

[2] Boiler Tests. George H. Barrus. Boston, 1891.

C. Air supplied through perforations in the top of the bridge wall and in the sides of the furnace after first passing through a series of passages running lengthwise of the walls of the setting.

D. Air admitted through perforated tiles in the sides of the furnace of a vertical tubular boiler, after having passed up and down through ducts in the walls.

E. Air admitted in similar manner in the case of a horizontal tubular boiler, after having passed through side-wall heating flues, as in the case of D.

Table No. 50.—Effect of Admitting Air above the Fire.

Designation.	Kind of Coal.	Gain.		Loss.	
		Per Pound of Coal.	Per Pound of Combustible	Per Pound of Coal.	Per Pound of Combustible
A	Cumberland,	5.9	6.2		
A	Anthracite, broken,			0.0	1.0
A	Two parts Pea and Dust and one part Cumberland,			2.0	4.7
B	Cumberland,	8.4	8.0		
B	Anthracite, broken,	1.9	3.7		
C	Two parts Pea and Dust and one part Culm,	2.0	0.0		
D	Two parts Anthracite Screenings and one part Cumberland,			4.3	2.3
E	Three parts Pea and Dust and one part Cumberland			4.5	4.7
F	Nova Scotia,	1.0	1.5		

F. Air supplied through perforations in the top of the bridge wall and inside of furnace.

Mr. Barrus concludes from these tests that "a considerable advantage attends the admission of air above the fuel when bituminous coal is employed, the amount of gain depending somewhat upon the method employed. There is no advantage in the system where mixtures of anthracite screenings and bituminous coal are used, if carried out according to either the first or fourth methods [A and C, as here designated], and finally, little or no benefit is derived when anthracite coal is burned."

It will be observed that four of the boilers here designated were provided with certain devices for heating the air before its admission to the fire. The gain in efficiency, by such an arrangement, is at the best but slight, and when the attempt is made to heat by such means all of the air supplied both below and above the fire, this plan is found to be totally inadequate. Such increase in temperature of the air supply as may be secured by proper and adequate appliances properly concerns the efficiency of the boiler, under which heading it will be discussed. Although it is undoubtedly true that heating the air intensifies the chemical affinities between the air and the fuel, it is doubtful if under ordinary conditions the effect is sufficient to be noticeable in boiler practice. This is, however, independent of the economy resulting from the abstraction of waste heat from the flue gases.

The best results from the admission of air above the fire appear to be secured when it is discharged in fine jets into the furnace chamber. For the accomplishment of such results other draft than that of the chimney is necessary. It is thus that mechanical draft becomes an important factor, both in thus furnishing the required air supply and in overcoming the added resistance which results from any attempt to preheat the air.

Loss on Account of Excess of Air. — As has already been stated, it is a practical impossibility so to distribute the amount of air theoretically required for the perfect combustion of a given fuel as to secure the ideal result. In practice, notwithstanding the fact that all excess means loss of efficiency of the fuel, it is necessary to supply enough additional air to ensure complete combustion even if a loss is occasioned thereby. What this excess should be, from an economical standpoint, can only be determined by practical experiment; and it will be found to vary with the character of the fuel, its rate of combustion, the temperature of the supply and the intensity of the draft.

For the purpose of illustration an anthracite coal may be taken, containing about 80 per cent of carbon, with an amount of inherent oxygen and hydrogen so small in quantity and of so little effect that, under the circumstances, it may be neglected. The amount of air required for the complete combustion of one pound of carbon having already been shown to be 11.3 pounds, that necessary for the combustion of one pound of this coal is, therefore, $11.3 \times 0.8 = 9.04$ pounds, as is also evident by the following calculation: —

Carbon,	0.8	Oxygen,	2.133
Oxygen,	2.133	Nitrogen,	6.906
Carbonic acid,	2.933	Air,	9.039
Nitrogen,	6.906	Carbon,	0.80
Products,	9.839	Products,	9.839

The specific heat of the products may be found, as follows, to be 0.2358.

	Weight.		Specific Heat.		B.T.U.
Carbonic acid . . .	2.933	×	0.2169	=	.6362
Nitrogen	6.906	×	0.2438	=	1.6837
	9.839				2.3199

$$\frac{2.3199}{9.839} = 0.2358$$

This value of the specific heat may be taken without appreciable error to apply to the products of combustion of carbon, no matter what the excess of air supplied. It is, therefore, evident that 1.0 × 0.2358 = 0.2358 pounds of water may be heated through 1 degree by the cooling of the gases through 1 degree.

By the methods here and previously described, Table No. 51 has been calculated for the purpose of showing clearly the effect upon the weight and ideal temperature of the gaseous products of combustion and their relative volume, resulting from the supply of air in various amounts in excess of that theoretically necessary for combustion.

Table No. 51. — Effects of Excess of Air.

Excess of Air, in per cent of Amount chemically required.	Total Weight of Air. Pounds.	Total Weight of Products of Perfect Combustion. Pounds.	Ideal Temperature above 62° in Heart of Fire. Degrees.	Relative Volume of Gases at Ideal Temperature above 62°.
0	9.04	9.84	5,053	1.00
50	13·56	14.36	3,461	1.04
75	15.82	16.62	2,491	1.06
100	18.08	18.88	2,633	1.09
125	20.34	21.14	2,351	1.11
150	22.60	23.40	2,124	1.13
175	24.86	25.66	1,937	1.15
200	27.12	27.92	1,780	1.17

Evidently, as the total volume of gases is heated in each case by the products of only one pound of coal, the total heat is constant; but because of the greater volume absorbing that heat the ideal temperature decreases as the air supply increases.

For simplicity, the effect of the greater specific gravity of carbonic acid has been neglected in the calculation, and the products of combustion have been taken as of constant density, at constant temperature, with varying amounts of air supplied.

The first effect of increasing the air supply is to lower the temperature of the products of combustion, as has already been shown and as is further indicated in Table No. 51. Were it not for the intervening boiler these gases would, therefore, pass to the chimney in increased volume but with decreased temperature. The absorption of heat by a boiler of the ordinary proportions serves, however, to bring about a result that is apparently paradoxical. In ordinary practice a boiler of good design is capable, under the usual conditions, of utilizing so much of the heat that the gases pass to the chimney at a temperature somewhere between 450° and 650°. An increase in the air supply to the furnace in connection with such a boiler actually results in raising the temperature of the escaping gases, as may be thus explained.

It is generally accepted that in the case of a steam boiler the amount of heat transmitted from the gases to the water, per degree difference between the gases and the boiler plates and tubes, is practically uniform for various differences of temperature. In other words, it is practically proportional to the difference in temperature. It is, further, evidently true that in the case of moving gases the amount of heat transmitted will be proportional to the time they remain in contact with the given surface. Owing to the lower temperature and greater density, their volume in the furnace is, therefore, not greatly increased by a moderate increase in the air supply, as is shown in Table No. 51.

But, as they pass across the heating surface and are cooled, their volume, and hence their velocity, decreases, and as their temperature approaches that of the admitted air so does their volume. In other words, the average rate of flow becomes more nearly proportional to the original air volume. As a consequence, the cooler gases resulting from the admission of a larger volume of air have less time in which to give up their heat, and are, therefore, cooled less in proportion to their temperature than are the hotter gases resulting from the admission of a smaller air volume. In addition, these cooler gases, because of the less difference between their temperature and that of the boiler, give up their heat less rapidly; the final result of these two influences, by which the transmission of heat is retarded, being that the gases accompanying the larger admission of air leave the boiler at a higher temperature than those which result from the admission of a smaller amount. Evidently, under these conditions the evaporative power of the boiler must be decreased.

M. Burnat conducted a series of experiments, showing that the temperature of the escaping gases increased with the supply of air. The boiler, provided with special heaters, had a heating surface of 475 square feet, with a grate of 18 square feet; the quantity of coal consumed was 293 pounds per hour, or 16 pounds per square foot of grate. These results are given in Table No. 52.

Table No. 52.—Effect of Increased Air Supply upon Temperature of Escaping Gases.

DAY.	Cubic Feet of Air at 62° per Pound of Coal.	Average Temperature of the Gases Leaving the Boiler.
First,	272	624°
Second,	198	601
Third,	168	550
Fourth,	124	487

The theoretical loss of efficiency, when air is supplied in excess and the products of combustion escape at different temperatures above the atmosphere, is exemplified in Table No. 53, the coal having a heat value of 11,720 B. T. U. This relates to the combustion of one pound of coal. An air supply 100 per cent in excess of that chemically required, and a temperature of escaping gases 450° above the atmosphere, represents fairly well the ordinary conditions of boiler practice with chimney draft, under which it will be noted that the loss is no less than 17.0 per cent. A chimney requires a high and wasteful temperature of gases to produce the draft, but this is unnecessary with mechanical means for draft production. A moderate reduction to 75 per cent excess air supply and 300° would show an economic gain of 7 per cent.

Table No. 53.—Loss of Efficiency Due to Excess of Air, and Temperature of Escaping Gases above Atmosphere.

Excess of Air in per cent of that chemically required.	Temperature of Escaping Gases above Atmosphere.											
	300°		350°		400°		450°		500°		550°	
	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.	Total B. T. U. in Gases.	Loss in per cent of Total Heat Value of Coal.
0	695	5.9	812	6.9	928	7.9	1,044	8.9	1,160	9.9	1,276	10.9
50	1,016	8.7	1,185	10.1	1,354	11.5	1,524	13.0	1,693	14.4	1,862	15.9
75	1,176	10.0	1,372	11.7	1,568	13.3	1,764	15.0	1,959	16.7	2,155	18.4
100	1,336	11.4	1,558	13.3	1,781	15.2	2,003	17.0	2,226	19.0	2,448	20.9
125	1,495	12.7	1,745	14.9	1,994	17.0	2,243	19.1	2,492	21.2	2,742	23.4
150	1,655	14.1	1,931	16.5	2,207	18.8	2,483	21.1	2,759	23.5	3,035	25.9
175	1,815	15.5	2,118	18.0	2,420	20.6	2,715	23.1	3,025	25.8	3,328	28.4
200	1,975	16.8	2,304	19.6	2,633	22.4	2,957	25.2	3,291	28.0	3,621	30.8

As bearing upon the evaporation of the boilers, the results embodied in Table No. 54 emphatically indicate the loss due to an increase in the air supply. These tests, by M. Burnat,[1] were upon boilers of various types. The air chemically required for perfect combustion was 130 cubic feet per pound in the case of the Ronchamp coal.

From these results it would appear that the best practice consisted in keep-

Table No. 54. — Effect of Air Supply on the Efficiency of Fuel.

BOILER.	COAL.	Relative Efficiency. Per cent	Coal per Hour. Pounds.	Air at 62° supplied per Pound of Coal. Cubic feet.	Water evaporated from and at 212° per Pound of Coal. Pounds.
Heating surface = 513 square feet. Grate surface = 14.2 square feet.	Ronchamp mixed.	100	330	219	7.09
		100	330	216	7.08
		107	330	174	7.62
		112	330	148	8.00
		113	330	127	8.06
Heating surface = 301 square feet.	Sarrebrück slack, very inferior.	100	284	222	5.46
		104	285	229	5.67
		108	276	200	5.93
		112	257	145	6.11
		112	242	153	6.13
		108	237	207	5.92
		110	234	126	6.01
Heating surface = 475 square feet. Grate surface = 18 square feet.	Sarrebrück slack, very inferior.	100	367	290	5.26
		108	370	264	5.67
		112	375	190	5.88
		114	361	141	6.03
		111	367	196	5.86
		101	316	121	5.32
Heating surface = 291 square feet.	Half Ronchamp slack and Sarrebrück slack.	100	280	190	6.60
		108	263	169	7.10
		107	259	152	7.09
		110	260	123	7.26

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[1] Bulletin de la Société Industrielle de Mulhouse, Vol. XXX. 1859–60.

ing the volume of air admitted as small as possible. But the result of such supply, although increasing the efficiency, even though imperfectly burning the coal, would cause a dull fire with abundant smoke and meet with serious practical objection on account of the difficulty of maintaining it under varying conditions of demand. It would, in fact, be a fire requiring the utmost care and attention on the part of the fireman, and liable to fail to raise steam when suddenly required.

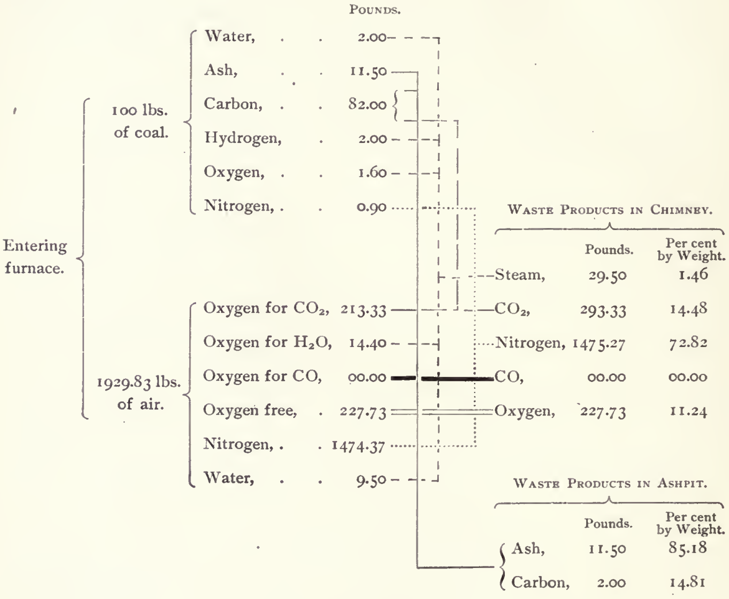
Summary of Influences Affecting the Efficiency of Fuel. — The relative importance of the principal influences which have just been discussed and the manner in which they are exerted is very clearly shown by an analysis of the results of combustion of a given quantity of coal. For illustration, there have been taken 100 pounds of anthracite coal of the composition shown in the accompanying tabular view. It is assumed that double the theoretical amount of air is supplied, that the atmosphere contains a normal amount of moisture, that two pounds of carbon remain unconsumed and pass to the ashpit, that neither carbonic oxide nor smoke is formed, that the coal and air have an original temperature of 60° when they enter the furnace, that the chimney gases have a temperature of 500°, and that the ash has a temperature of 450°.

The total heat of 100 pounds of the fuel is shown to be 1,313,080 B. T. U., but, owing to direct loss to the ashpit, the heat generated amounts to only 1,283,780 B. T. U. The Tabular View indicates the relative weights of the various products of combustion, and the accompanying Table No. 55 shows the heat losses from these sources, together with their method of calculation. It is

Table No. 55. — Heat Losses Incident to the Combustion of 100 Pounds Anthracite Coal.

Heat Losses.	Number of B. T. U.	Per cent of Total Heat of Fuel.
By water = [(212 − 60) × wt.] + 965.7 × wt. + [sp. heat × (500 − 212) × wt.],	37,012.5	2.83
By carbonic acid = wt. × sp. heat × (500 − 60),	27,994.2	2.13
By nitrogen = wt. × sp. heat × (500 − 60),	158,452.8	12.07
By free oxygen = wt. × sp. heat × (500 − 60),	21,973.6	1.67
By ash = wt. × sp. heat × (450 − 60),	1,105.7	0.08
By carbon in ash = wt. × sp. heat × (450 − 60) + wt. × 14,650,	29,488.3	2.24
By carbonic oxide = wt. × sp. heat × (500 − 60) + wt. × 4,400,	——	——
Total heat lost exclusive of loss by radiation,	276,027.1	21.02
Theoretically possible evaporation in pounds of water from and at 212° per pound of combustible utilized,	12.73	
Theoretically possible evaporation in pounds of water from and at 212° per pound of fuel utilized,	10.44	

Tabular View Showing Results of Combustion of 100 Pounds of Anthracite Coal with Twice the Theoretical Amount of Air Required.



Entering furnace.			Pounds.
	100 lbs. of coal.	Water,	2.00
		Ash,	11.50
		Carbon,	82.00
		Hydrogen,	2.00
		Oxygen,	1.60
		Nitrogen,	0.90
	1929.83 lbs. of air.	Oxygen for CO_2,	213.33
		Oxygen for H_2O,	14.40
		Oxygen for CO,	00.00
		Oxygen free,	227.73
		Nitrogen,	1474.37
		Water,	9.50

Waste Products in Chimney.

	Pounds.	Per cent by Weight.
Steam,	29.50	1.46
CO_2,	293.33	14.48
Nitrogen,	1475.27	72.82
CO,	00.00	00.00
Oxygen,	227.73	11.24

Waste Products in Ashpit.

	Pounds.	Per cent by Weight.
Ash,	11.50	85.18
Carbon,	2.00	14.81

Total heat of fuel.

$$\left\{\begin{array}{ll} \text{Weight of C} \times 14{,}650 = 82 \times 14{,}650 = \quad . \quad . \quad . & 1{,}201{,}300 \text{ B. T. U.} \\ \text{Wt. of H} - \left(\frac{\text{Wt. of O}}{8}\right) \times 62{,}100 = 2 - \left(\frac{1.6}{8}\right) \times 62{,}100 = & \underline{111{,}780} \quad \text{"} \\ & 1{,}313{,}080 \quad \text{"} \end{array}\right.$$

Heat generated.

$$\left\{\begin{array}{ll} 80 \times 14{,}650 = \quad . \quad . \quad . \quad . \quad . \quad . \quad . & 1{,}172{,}000 \text{ B. T. U.} \\ 2 - \left(\frac{1.6}{8}\right) \times 62{,}100 = \quad . \quad . \quad . \quad . \quad . \quad . & \underline{111{,}780} \quad \text{"} \\ & 1{,}283{,}780 \quad \text{"} \end{array}\right.$$

evident that under the assumed conditions 21.02 per cent is irretrievably lost, and that neglecting the loss by radiation from brickwork, the efficiency of the fuel is 78.98 per cent. With the given fuel these losses can only be lessened by decreasing the air supply, preventing the loss of some of the carbon to the ashpit, and by proper means lowering the final temperature of the escaping gases. The total loss through moisture in coal and air, ash in coal, and carbon in ash amounts to 5.15 per cent.

Commercial Efficiency of Coals. — The cost of producing a given amount of steam is the ultimate criterion by which the efficiency of any fuel must be judged. The commercial efficiency not only concerns the amount of water evaporated by a pound of fuel, but is directly dependent upon the following items of expense: *first;* interest, rent, taxes and insurance on the cost, and the depreciation of the plant; *second*, repairs; *third*, cost of water from which the steam is generated; *fourth*, labor; *fifth*, getting rid of the ashes; *sixth*, cost of the fuel in the boiler house.

The cost of water may be considered practically constant in the comparison of fuels in a given boiler plant, but the items of interest, depreciation and repairs are to a great extent directly dependent upon the fuel; for with the cheaper fuels more expensive contrivances are usually necessary, and under the same conditions the output of the plant decreases with the quality of the fuel. This is quite clearly shown in Table No. 46, in which the water evaporated per square foot of heating surface grows less as the quality of the fuel degenerates. It is still further shown by Table No. 56, in which the horse-power of various

Table No. 56. — Effect of Quality of Fuel upon Output of Boiler.

KIND OF FUEL.	RATED HORSE-POWER OF BOILER.					
	54	74	87	129	140	270
Cumberland,	60.0	143.8	105.4		94.0	214.6
Anthracite, broken,	53.9	105.5	84.0	192.3	103.8	196.1
Pea,				149.2		
One part Screenings and one part Cumberland,		95.1				
Three parts Screenings and one part Cumberland,						204.8
Two parts Pea and Dust and one part Cumberland,	38.5		82.2		118.1	
Forty-four parts Pea and Dust and thirty-seven parts Culm,				157.1		

boilers, tested by Mr. George H. Barrus[1] under somewhat different draft conditions, is compared with their rated power when using various kinds of coal. This decreased power is directly due to the reduced combustion per square foot of grate, which may be counteracted to a certain extent by increasing the grate surface.

The largest and most important factor, however, is the cost of the fuel itself, which should be measured not by the number of pounds but by the available heat units obtained for a given price. In this cost are properly included the transportation charges, the expense of getting the coal into the boiler house and putting it into the furnace as well as taking out and carrying away the ash. Practically all of these costs are directly dependent upon the weight of the coal, regardless of the number of heat units it is capable of developing. But the net cost depends not upon the number of units in the coal but upon the number that can be utilized under the given conditions. The haulage of ashes becomes so important in some cases that it is found more economical to pay a higher price for a coal containing less ash rather than go to the necessary expense of teaming the ashes a considerable distance.

As the cost of transportation of the coal is practically dependent upon the weight and independent of the character of the coal, the proportional difference in price which may rule at the mine may be almost extinguished when an equal charge is added for the transportation of each. Thus No. 2 buckwheat may prove a very economical fuel when utilized at the mine, where it may be purchased at 25 cents per ton, although its direct calorific efficiency is low and it requires a special form of furnace for satisfactory combustion. No. 1 buckwheat coal costing 50 to 60 cents under the same conditions may, however, prove to be a close rival because of its higher efficiency and the greater ease with which it may be consumed. But this difference of 100 to 120 per cent in cost becomes reduced to a difference of only 11 to 15 per cent when a transportation charge of $2.00 per ton is added to each. Both of these coals, although requiring a larger boiler plant for the same aggregate evaporative results, may, after transportation charges are added, prove on the whole more economical than larger sized anthracite of higher efficiency but at higher prices.

A comparison of the losses shown in Table No. 55, reduced to the commercial relation when different kinds of coal are used, serves to make the preceding clear. This comparison is presented in Table No. 57, showing the relation between an anthracite buckwheat costing 50 cents per ton at the mines, the same coal when transportation charges of $2.00 have been added, and pea coal costing $1.50 at the mines, with the same expense for transportation as the buckwheat. In each case the coal is assumed to be burned with double the theoretical amount of air, while the loss by radiation, which might amount to from 4 to 20 per cent, has not been considered This loss, if known, should be deducted from the amount here given as actually utilized. The cost is based in the four cases upon the theoretical evaporation of 10.44 pounds of water from and at 212° per pound of coal, as given in Table No. 55. In practice these results would undoubtedly have to be corrected, relatively as well as directly, because of the higher evaporation probable with the pea coal.

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[1] Boiler Tests. George H. Barrus. Boston, 1891.

Table No. 57.—Commercial Value of the Losses Incident to Burning 100 Pounds of Coal.

LOSSES.	KIND AND PRICE OF COAL.			
	Buckwheat Coal.		Pea Coal.	
	$0.50	$2.50	$1.50	$3.50
By water,	$0.00063	$0.00315	$0.00189	$0.00442
By carbonic acid,	0.00048	0.00238	0.00143	0.00333
By nitrogen,	0.00269	0.01347	0.00808	0.01886
By free oxygen,	0.00037	0.00186	0.00112	0.00261
By ash,	0.00002	0.0009	0.00005	0.00012
By carbon in ash,	0.00050	0.0025	0.00150	0.00350
Total loss, not including radiation from brickwork,	0.00469	0.02245	0.01407	0.03284
Actually utilized, including radiation from brickwork,	0.01763	0.08816	0.05289	0.12341
Cost of fuel to evaporate 100 pounds of water from and at 212°,	0.00214	0.01069	0.00641	0.01497

Based upon the cost of the coal and firing, the following will serve to illustrate the economy that may be secured by burning a cheaper fuel of lower efficiency. In a certain plant equipped with horizontal tubular boilers and down-draft furnaces, an evaporation of 8.5 pounds of water from and at 212° was regularly obtained. The coal used was Illinois slack, costing $1.40, delivered under the boilers. This makes the cost about 8.3 cents per 1,000 pounds evaporated under these conditions. Had the best picked anthracite coal, costing $4.80 per ton, under existing market rates, been used, and had it evaporated the generous amount of 12 pounds of water from and at 212°, the cost would have been 20 cents per 1,000 pounds.

Taking the relative values given in Table No. 43 and the prices ruling at the time and in the locality of the tests, and following Mr. Barrus' method of comparison, the cost of coal of the various kinds necessary to generate 1,000 horse-power may be calculated. A horse-power, as applied as a standard of boiler capacity, is elsewhere explained as being equivalent to the evaporation of 34.488 pounds of water from and at 212°. The daily production of steam during a ten hours' run of a 1,000 horse-power plant would, therefore, be 1,000 × 10 × 34.488 = 344,880 pounds from and at 212°. The results of calculation thus obtained are embodied in Table No. 58, as well as those showing the cost of labor and the total cost of coal and labor. In estimating the labor the assumption is made that in the case of anthracite coal of broken, chestnut and pea sizes the labor is performed by two day firemen, one night fireman and two helpers, and in the case of the bituminous coal one additional fireman is required; while for the mixed coals one fireman and one helper additional are necessary. It is further assumed that the wages of the firemen are $1.75 per day and of the helpers $1.25 per day. From this table it is evident that under the prevailing prices there is practically no choice between the Cumberland, anthracite, pea, and mixture of pea and dust with Cumberland; while the Nova Scotia culm and the other mixture do not fall far behind in efficiency. The anthracite broken and chestnut coals are, however, not only most expensive per ton, but are also the least efficient from a commercial standpoint. The

Table No. 58.—Cost of Coal and Labor for a Day's Run of Ten Hours, 1,000 Horse-power Plant.

	Cumberland.	Anthracite, Broken.	Anthracite, Chestnut.	Two parts Pea and Dust and one part Cumberland.	Two parts Pea and Dust and one part Culm.	Anthracite, Pea.	Nova Scotia Culm.
Weight of coal used in ten hours (2,240 lbs. = 1 ton), tons,	13.9	15.7	16.4	16.4	17.1	17.4	18.3
Cost of coal per ton of 2,240 lbs., $	4.56	5.65	6.13	3.72	3.29	3.74	3.28
Cost of coal used in ten hours, $	63.38	88.70	100.53	61.05	56.26	62.25	60.02
Cost of labor per day, $	9.50	7.75	7.75	10.75	10.75	7.75	9.50
Total cost of coal and labor per day, $	72.88	96.45	108.28	71.80	67.01	73.00	69.52
Relative practical heat value (Cumberland = 100), per cent,	100	89	85	85	82	80	76
Relative cost of coal per ton (Cumberland = 100), per cent,	100	124	134	82	72	81	72
Relative total cost of coal and labor per day (Cumberland = 100), per cent,	100	132	149	99	92	100	95

relative values in per cent — based on Cumberland as 100 — serve to show clearly the stated relations existing between the different coals. Based upon the cost of the coal alone, the rate of evaporation must vary inversely as the cost in order to secure equivalent results in water evaporated per unit of cost. This principle has been carried out in the calculations of Table No. 59. Thus, if coal at $3.50 per ton has an evaporative power of unity, it will be necessary for one pound of coal at $5.00 per ton to evaporate 1.43 times as much water to produce the same commercial result; or if coal costing $1.50 per ton is substituted for coal costing $4.00 per ton, the cost per pound of water evaporated will be the same if the latter fuel evaporates 0.38 as much as the former.

Table No. 59. — Rates of Evaporation for Equivalent Cost of Coal.

Cost of Coal per Ton.	Cost of Coal per Ton.													
	$0.50	$1.00	$1.50	$2.00	$2.50	$3.00	$3.50	$4.00	$4.50	$5.00	$5.50	$6.00	$6.50	$7.00
$0.50	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00	12.00	13.00	14.00
1.00	0.50	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00	6.50	7.00
1.50	0.33	0.67	1.00	1.33	1.67	2.00	2.33	2.67	3.00	3.33	3.67	4.00	4.33	4.67
2.00	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50
2.50	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	2.00	2.20	2.40	2.60	2.80
3.00	0.17	0.33	0.50	0.67	0.84	1.00	1.17	1.33	1.50	1.67	1.83	2.00	2.17	2.33
3.50	0.14	0.29	0.43	0.57	0.71	0.86	1.00	1.14	1.28	1.43	1.57	1.71	1.86	2.00
4.00	0.13	0.25	0.38	0.50	0.63	0.75	0.88	1.00	1.13	1.25	1.38	1.50	1.63	1.75
4.50	0.11	0.22	0.33	0.45	0.56	0.67	0.78	0.89	1.00	1.11	1.22	1.33	1.44	1.55
5.00	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00	1.10	1.20	1.30	1.40
5.50	0.09	0.18	0.28	0.36	0.45	0.55	0.64	0.73	0.82	0.91	1.00	1.09	1.18	1.27
6.00	0.08	0.17	0.25	0.33	0.42	0.50	0.58	0.67	0.75	0.83	0.92	1.00	1.08	1.17
6.50	0.07	0.15	0.23	0.31	0.39	0.46	0.54	0.62	0.69	0.77	0.85	0.92	1.00	1.08
7.00	0.07	0.14	0.21	0.29	0.36	0.43	0.50	0.57	0.64	0.71	0.79	0.86	0.93	1.00

Influence of Mechanical Draft. — Intense draft is one of the most important factors in the utilization of cheap fuels; hence the value of mechanical draft. Its influence will be discussed later at length, but it is here proper to present at least one illustration of the increased economy in fuel resulting from its use.

At the United States Cotton Company's mills, at Central Falls, R. I., is a boiler plant consisting of 3 Babcock & Wilcox boilers of 335 horse-power each,— a total of 1,005 horse-power. The draft is furnished by a Sturtevant blower, which forces the air to the ashpits, and whose speed is automatically regulated by a special device, so that the volume of air and intensity of draft are continually changing to suit the varied conditions and requirements of the fire. Before this mechanical draft plant was put into operation, the fuel employed was George's Creek Cumberland coal, costing $4.00 per ton of 2,200 pounds, delivered at the boiler room. Since the fan with automatic control has been in use, the quality and price of the fuel has been reduced to a mixture of about 70 to 75 per cent of No. 2 buckwheat, 20 to 25 per cent of yard screenings, and 5 to 10 per cent of Cumberland, costing $2.62 per ton. This change has reduced the fuel cost per indicated horse-power on the engine by $0.001235 per horse-power per hour. At the horse-power developed by the engines, the present saving per week is over $126.00, when all fuel, including that for banking fires, is taken into account. Comment is unnecessary.

Prevention of Smoke. — The tendency of a coal to produce smoke increases with the volatile combustible matter which enters into its composition. Pure carbon and coke are smokeless, and the best anthracite coal is practically so, but the bituminous coals are as a rule distinguished for their smoke-producing qualities. As has already been shown, the actual amount of unconsumed carbon passing away from a well-constructed boiler in the form of smoke seldom if ever exceeds one per cent of the total amount of carbon in the coal. Nevertheless, because of the visible effect of even such a small amount, the "smoke nuisance" is widespread and of serious consequence in some localities. Many have been the devices presented and applied for overcoming this evil, although but few have met with ultimate success.

As smoke is a result of incomplete combustion, its prevention must be sought through the provision of an ample supply of air, with sufficient intensity of draft and the maintenance of a high temperature of the fuel bed. Unless the proposed preventative device meets these requirements it has little hope of success.

The contrivances which have been applied for the purpose may be broadly classed as follows: —

I. Mechanical or forced draft.

II. Arrangements for admission of air above the fire, under which may be included steam jets for inducing a flow of air.

III. Firebrick arches or checker work, placed over the bridge wall or near the end of the fireplace, for the purpose of mixing and heating the gases.

IV. Hollow walls for preheating air.

V. Coking arches or chambers constructed in front of the fireplace, whence the coke is pushed to the rear as the volatile matter is distilled off.

VI. Double combustion, whereby part or all of the gases are passed a second time through the fuel.

VII. Down-draft furnaces in which air is admitted above the grate and the gases pass down through it and thence to the heating surface.

VIII. Automatic stokers.

With coals of moderate smoking qualities mechanical draft in its simplest application, by its power to furnish an adequate amount of air under a pressure sufficient to cause it to pass readily through the fuel, meets all the requirements of a smoke preventative, even when all of the air is admitted beneath the grate. With excessively smoky coals, however, a portion of the air may be omitted above the fuel. The best results are obtained when this air enters the furnace under the influence of a positive means like mechanical draft, in a series of jets, by which it is forced to commingle with the gases as they rise from the bed of fuel. Owing to the tendency of cold air thus admitted to chill the fire and actually increase the amount of smoke, it is desirable, when rapid combustion takes place, that the air be preheated to a considerable degree.

The steam jet has also been employed to induce a flow of air which, mixed with steam and thus heated, is forced either beneath or above the fire, as the case may be. While the admixture of steam incident to this method of draft production lessens the tendency to clinker, nevertheless its cost of operation is much greater than that of the fan blower, as will be shown later. Evidently all of the steam thus admitted must be raised to the temperature of the escaping gases, thereby reducing the efficiency of the fuel.

Mr. J. C. Hoadley[1] made clear the inefficiency of preheating devices where the draft is not sufficiently strong to cause the necessary movement of air, and the inadequacy of passages in brick settings as means of heating the air prior to its admission above the fire. In the boiler under test these passages were formed in the brick setting of the back and sides, in the rear of the bridge wall, and communicated with openings at the bridge wall and in the side walls of the furnace. Sliding dampers were provided to regulate the admission of air; but, as stated by Mr. Hoadley, "careful and repeated experiments and observations proved that these dampers could never be opened without checking the draft through the fuel and lowering the temperature of the fire; and it is not impossible that a very slight leakage through the closed dampers may have lowered the efficiency of the boilers."

As regards the arrangement "for heating air or 'superheating it' (whatever superheating may be supposed to mean when applied to a permanent gas), . . . no good was ever found to result from this system of flues; indeed it is doubtful if any considerable quantity of air ever passed through the flues at all, although some must have flowed in when the dampers were opened, since the resistance of the open flue, circuitous as it was, could hardly have been so great as that of the coal on the fire grates."

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

With a temperature of about 2,000° immediately above the fire in the ordinary furnace, it is evident that any device which heats the air but a few degrees above the temperature of the atmosphere can have no practical effect upon the combustion, and that some more elaborate arrangement is necessary. As such devices pertain more properly to the efficiency of the boiler, they will be discussed under that head.

Firebrick arches and coking chambers are both serviceable in preventing the formation of smoke, but require very careful management.

As the inflammable gas (CO) and the unconsumed carbon together seldom exceed two per cent of the total gases, any attempt to burn them again, as by "double combustion," is futile. Apparent success is due to the admission of additional air.

The down-draft furnace, owing to reversal of the usual direction of movement of the gases, requires a water grate to withstand the intense heat. The practical features of such construction generally make it impossible to prevent considerable of the fuel dropping through the grate. To avoid loss from this source an auxiliary grate is usually provided, upon which this fuel may be consumed. Even with a forced fire and careless firing such an arrangement appears to insure a good smoke record. The conditions are such, however, that greater draft is required than with the ordinary type of furnace, so that mechanical draft is often found to be of especial importance.

As a rule, automatic stokers are introduced for reasons other than the prevention of smoke, although, by their uniformity of feeding and their frequent application of the coking principle in their construction, they are capable, under favorable conditions, of giving good results. They do not, however, readily handle caking or clinkering coals, and usually require coal of the large sizes, and hence are restricted in their application.

With ordinary coal and hand-firing the prevention of smoke is largely dependent upon the fireman, irrespective of any special appliances; for these, no matter how excellent their character, are in course of time likely to be neglected. Many devices applied for this purpose, or for increasing the efficiency, have shown favorable results merely because they compelled greater attention on the part of the fireman in the care of boilers that were previously worked with marked inefficiency. Beyond certain factors, such as a sufficiency of draft, over which he can have no control, a good fireman is, after all, the most important factor in increasing efficiency and preventing smoke.

CHAPTER VI.

EFFICIENCY OF STEAM BOILERS.

Measure of Efficiency. — The practical efficiency of a boiler and that of the fuel consumed in connection with it are interdependent. That is, the attainable efficiency of the fuel is dependent upon the design and operation of the boiler; while the efficiency of a given boiler is a direct measure of the amount of heat derived from the fuel employed. It has long been the custom to compare boilers upon the basis of the number of pounds of water evaporated by each per pound of fuel or combustible burned. Of course, efficiencies measured thus vary greatly in different types of boilers; but, broadly stated, it is undoubtedly true that the rate of evaporation per pound of coal from feed-water at 60° into steam of 80 pounds gauge pressure is in general below 8 pounds. This is equivalent to 9.56 pounds from and at 212°. Indeed, 8 pounds of dry steam under the above conditions is a fair result, 8.25 pounds a good result, 8.5 pounds very good and 9 pounds about the best usually attainable. This latter amount corresponds to 10.74 pounds from and at 212°, is equivalent to 69 per cent of the full calorific power of carbon, and is for coal of five-sixths carbon a high result.

Results thus compared are, however, liable to be deceptive,— in some cases intentionally so,— because of the wide variation in the quality of coal. Thus, by means of picked coal of high calorific power, an exceptionally high evaporation may be obtained and used to advocate the merits of a given boiler. Less attention is generally given to the quality of the coal than to the amount of water evaporated per pound, and the fact is not always recognized that a poor boiler tested with good coal may actually give a greater evaporation than a good boiler with poor coal. The possibility of obtaining such results is rendered evident by Table No. 60, showing three sets of assumed but perfectly practical conditions.

If the measure of efficiency were to be based solely upon the evaporation per pound of coal, boiler B would be selected. But it is equalled in efficiency by boiler C, if the efficiency be measured by the proportion of available heat utilized, although it evaporates 1.33 pounds less water because of the poorer quality of the coal. Whether it would be commercially the more efficient of the two would depend upon the relative cost of the two kinds of coal, and that having a heat value of 9,500 B. T. U. would have to cost about 14 per cent less. Based upon water evaporated, boiler A appears more efficient than boiler C, but measured by the amount of heat absorbed the latter far exceeds the former. Although boiler A, under the given conditions, evaporates about 9 per cent more than boiler C, boiler C is the more economical in the combustion of coal by about 40 per cent.

Table No. 60. — Relative Efficiency of Boilers.

Designation of Boiler.	Heat Value of One Pound of Coal. B. T. U.	Evaporation from and at 212° per Pound of Coal. Pounds.	Efficiency. Per cent.
A	14,500	7.50	50
B	11,250	8.19	70
C	9,500	6.86	70

It is, therefore, evident that the evaporation from and at 212°, although a very convenient basis of comparison for fuels, may not be properly applied in defining the efficiency of a boiler. It is thus that the distinction is to be drawn between the efficiency of a fuel and that of the boiler in connection with which it is burned. A more accurate basis for the comparison of efficiency of different types of boilers is established when the evaporation is expressed in pounds of water evaporated per pound of combustible. But even this is somewhat affected by the proportion of ash and elementary water in the coal.

The ideal basis of comparison is to be sought in the ratio found by experiment to exist between the total effective heat of the coal, as determined by means of a calorimeter, and that rendered evident in the steam generated, which may be thus expressed : —

$$\text{Efficiency} = \frac{\text{Heat units usefully applied.}}{\text{Heat units supplied to furnace.}}$$

Although ideally the correct method of comparing the efficiency of boilers, and likewise of fuels, when proper allowances can be made for boiler differences, it is practically open to criticism because of the difficulty in determining experimentally from a small specimen the exact heating value of the entire quantity of coal. Even with this basis of comparison, relative efficiency tests should be conducted under identical conditions so far as they may be obtainable. The complete results can be best presented in the form of a heat balance, as follows : —

Heat Balance.

Dr.	*Cr.*
To heat	By heat
from coal,	in dry steam,
from air,	in moisture and water mechanically suspended in steam,
from feed-water.	in dry flue gases,
	in moisture in coal, in water resulting from combustion, in vapor in air, } at temperature of flue gases.
	lost through incomplete combustion to CO,
	in ashes,
	lost by radiation and otherwise unaccounted for.

As is the case with fuel, so is it with a boiler: the efficiency must be considered commercially. For this reason a limit is reached considerably short of 100 per cent, beyond which the loss in interest, depreciation and other fixed charges exceeds the gain from decreased cost of fuel, per unit of evaporation, resulting from the given improvement.

In illustration of the difference in efficiency of different types of boilers, Table No. 61 is presented. This covers the results of 86 tests conducted by Mr. Wm. H. Bryan[1] under common conditions and with ordinary fuel, principally Illinois coal. Due allowance is to be made for locality, special type of boiler and kind of coal.

Table No. 61.—Efficiency of Different Types of Boilers.

KIND OF BOILER.	Number of Trials.	EFFICIENCY IN PER CENT.		
		Maximum.	Minimum.	Average.
Small vertical,	3	46.10	34.60	41.60
Large vertical,	3	52.30	49.00	50.90
Large vertical, improved setting,	1	——	One trial only.	67.89
Tubular boilers,	14	60.17	44.76	51.53
Tubular boilers, improved settings,	34	76.38	41.94	58.87
Water-tube boilers,	13	70.11	49.37	61.31
Water-tube boilers, improved setting,	18	81.32	49.30	67.52

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[1] Boiler Efficiency, Capacity, and Smokelessness, with Low-Grade Fuels. Wm. H. Bryan. A paper read before the Engineers' Club of St. Louis, Oct. 21, 1896.

Rating of Steam Boilers. — As originally the most important purpose of the steam boiler was to generate steam for use in a steam engine, it became customary to express its capacity by the nominal output of the engine which it supplied. That is, its rating was expressed in horse-power (a term with which the boiler has properly nothing to do); a horse-power in each case representing the weight of steam required per hour to enable the engine to perform work continuously at the rate of 33,000 foot-pounds per minute. The first standard, fixed by Watt, and based upon the performance of the engines of his day, was one cubic foot of water (weighing about 60 pounds) evaporated per hour from 212° per horse-power.

As the efficiency of the steam engine has been improved, the amount of steam necessary for the production of a horse-power has been gradually decreased. The attempt was made to keep pace with engine improvements by correspondingly reducing the amount of water represented by a horse-power. But the present existence of engines, in great variety of design and manner of operation, as compared with the simple type in the days of Watt, renders impossible the establishment of any definite standard of rating which shall apply to them all. This cannot be more clearly evidenced than by Table No. 62, which

Table No. 62. — Steam per Horse-Power per Hour for Steam Engines with Different Ratios of Expansion.

Type of Engine.	Steam Pressure above Atmosphere	RATIO OF EXPANSION.					
		2	3	4	5	7	10
Non-condensing,	30	40	39	40	40	42	45
	45	35	34	36	36	38	40
	60	30	28	27	26	30	32
	75	28	27	26	25	27	29
	90	26	25	24	23	25	27
	105	25	24	23	22	22	21
	135	24	23	22	21	20	20
Condensing,	15	30	28	28	30	35	40
	30	28	27	27	26	28	32
	45	27	26	25	24	25	27
	60	26	25	25	23	22	24
	75	26	24	24	22	21	20
	105	25	23	23	22	21	20
	135	25	23	22	21	20	19

embodies Prof. R. H. Thurston's[1] estimate of the steam consumption of the best classes of engines in common use and in good order. Evidently the term "horse-power," applied to the rating of steam boilers, must, therefore, be considered as a standard of measurement rather than a direct measure of capacity.

In 1876 the committee of judges of the Centennial Exhibition, to whom was entrusted the trials of the competing boilers exhibited, decided to adopt a standard rating upon conditions considered by them to represent fairly average practice. The standard unit of 30 pounds of water of 100° temperature evaporated into dry steam of 70 pounds gauge pressure, thus adopted, has since become the almost universal standard of rating in the United States for the nominal evaporative capacity of steam boilers, and is commonly designated as a commercial horse-power. The amount of heat required to evaporate one pound of water under these conditions is 1,110.2 B. T. U., equivalent to 1.1496 units of evaporation, as previously defined. An evaporation of 30 pounds under the stated condition is, therefore, equivalent to the development of 33,305 B. T. U. per hour, or 34.488 pounds of water evaporated from and at 212°, or in round numbers 34.5 pounds. Although even this rating is now open to criticism, because it does not represent the present standard of average steam-engine performance, nevertheless, being a more or less arbitrary standard, it would appear to be as satisfactory as any other for the mere purpose of comparing the capacity of different boilers. Any standard of this character is, however, open to the criticism that because of the range of capacity possessed by any boiler it is difficult to fix the conditions under which this capacity should be attained. Should they be those that exist when the boiler is being run easily under ordinary conditions, or should the measure of capacity be taken when the boiler is pushed to its utmost? The above-mentioned committee considered that a boiler should be capable of developing its stipulated power with easy firing, moderate draft and ordinary fuel, while exhibiting good economy; and that the boiler should, furthermore, be capable of developing at least one-third more than its rated power to meet emergencies. It is obvious, nevertheless, that no matter what the basis adopted, a rating based upon the horse-power standard possesses considerable elasticity.

Table No. 63 is presented for the purpose of simplifying the reduction of horse-power under any given conditions to that under other conditions.

The basis is 30 pounds of water at 100° evaporated into steam of 70 pounds pressure. At any other temperature, as for instance 170°, and pressure of say 80 pounds, the equivalent evaporation is shown to be 31.96 pounds.

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[1] A Manual of Steam Boilers. R. H. Thurston. New York, 1888.

Table No. 63. — Required Hourly Evaporation per Commercial Horse-Power at Various Temperatures of Feed and Pressure of Steam.

Temperature of Feed-Water.	STEAM PRESSURE ABOVE THE ATMOSPHERE.																				
Degrees Fahr.	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200
50	29.51	29.29	29.14	29.02	28.92	28.84	28.77	28.70	28.64	28.59	28.54	28.49	28.45	28.41	28.37	28.33	28.29	28.26	28.23	28.20	28.17
60	29.77	29.55	29.40	29.28	29.18	29.09	29.02	28.95	28.89	28.84	28.79	28.74	28.69	28.65	28.61	28.57	28.54	28.51	28.48	28.45	28.42
70	30.04	29.81	29.66	29.54	29.44	29.35	29.27	29.21	29.15	29.09	29.04	28.99	28.94	28.90	28.86	28.82	28.78	28.75	28.72	28.69	28.66
80	30.31	30.08	29.93	29.80	29.70	29.61	29.53	29.46	29.40	29.34	29.29	29.24	29.19	29.15	29.11	29.07	29.03	29.00	28.97	28.94	28.91
90	30.59	30.36	30.20	30.07	29.97	29.88	29.80	29.73	29.67	29.61	29.55	29.50	29.45	29.41	29.37	29.33	29.29	29.25	29.22	29.19	29.16
100	30.88	30.64	30.47	30.34	30.24	30.15	30.07	30.00	29.93	29.87	29.82	29.77	29.72	29.67	29.63	29.59	29.55	29.51	29.48	29.45	29.42
110	31.17	30.93	30.76	30.63	30.52	30.43	30.34	30.27	30.20	30.14	30.09	30.04	29.99	29.94	29.90	29.86	29.82	29.78	29.74	29.71	29.68
120	31.46	31.22	31.05	30.91	30.80	30.71	30.63	30.55	30.48	30.42	30.36	30.31	30.26	30.21	30.17	30.13	30.09	30.05	30.01	29.98	29.95
130	31.76	31.52	31.34	31.20	31.09	30.99	30.91	30.83	30.76	30.70	30.65	30.59	30.54	30.49	30.45	30.41	30.37	30.33	30.29	30.25	30.22
140	32.07	31.82	31.64	31.50	31.38	31.29	31.20	31.12	31.05	30.99	30.93	30.88	30.83	30.78	30.73	30.69	30.65	30.61	30.57	30.53	30.50
150	32.39	32.12	31.94	31.80	31.68	31.58	31.50	31.42	31.35	31.28	31.22	31.17	31.12	31.07	31.02	30.97	30.93	30.89	30.85	30.81	30.78
160	32.71	32.44	32.26	32.11	31.99	31.89	31.80	31.72	31.65	31.58	31.52	31.46	31.41	31.36	31.31	31.27	31.23	31.19	31.15	31.11	31.08
170	33.03	32.76	32.58	32.43	32.31	32.20	32.11	32.03	31.96	31.89	31.83	31.77	31.71	31.66	31.61	31.56	31.52	31.48	31.44	31.40	31.37
180	33.37	33.09	32.90	32.75	32.63	32.52	32.43	32.34	32.27	32.20	32.14	32.08	32.02	31.97	31.92	31.87	31.83	31.79	31.75	31.71	31.67
190	33.71	33.43	33.23	33.08	32.95	32.84	32.75	32.66	32.59	32.50	32.45	32.39	32.33	32.28	32.23	32.18	32.14	32.10	32.06	32.02	31.98
200	34.06	33.77	33.57	33.41	33.28	33.17	33.08	32.99	32.91	32.84	32.77	32.71	32.65	32.60	32.55	32.50	32.45	32.41	32.37	32.33	32.29
212	34.49	34.18	33.98	33.80	33.69	33.58	33.48	33.39	33.31	33.24	33.17	33.11	33.05	32.99	32.94	32.89	32.84	32.80	32.76	32.72	32.68

The preceding applies most directly to the rating of boilers as determined by experimental test. But it is obviously desirable that, for the purposes of designation, the capacity or rating of a given design of boiler should be at least approximately known before it is constructed. Evidently such a rating must be based upon previous experiment with similar boilers, having the same general proportions and disposition of heating surface and operating under similar conditions. The measure of such results is made in the number of pounds of water evaporated per hour per square foot of heating surface, or its equivalent, the number of square feet of heating surface per commercial horse-power. In boilers of the same relative proportions of grate to heating surface, and of the same general arrangement of heating surface, the rate of evaporation is fairly constant for ordinary draft conditions. That is, the total evaporation is practically proportional to the heating surface. The area of this surface, therefore, forms a ready means of comparison of capacity. But the location and character of the heating surface largely determines the rate of evaporation, thus restricting direct comparison to boilers of the same type under the same conditions. A fair average for boilers of the ordinary horizontal return tubular type, with ordinary natural draft, is an evaporation of 3 pounds of water from and at 212° per square foot of heating surface. On the basis of 34.5 pounds per horse-power, under these conditions, this is equivalent to one horse-power for each 11.5 square feet of heating surface. The number of square feet of heating surface per horse-power, for other rates of evaporation, is presented in Table No. 64.

Table No. 64. — Square Feet of Heating Surface per Horse-Power.

Pounds of water evaporated from and at 212° per square foot of heating surface per hour,	2.0	2.5	3.0	3.5	4.0	5.0	6.0	7.0	8.0	9.0	10.
Square feet of heating surface required per horse-power,	17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8	3.5

Under mechanical draft the rate of evaporation per unit of heating surface is greatly increased. Thus "the maximum power obtained with forced draft and an air pressure not exceeding 2 inches of water," as stated by Sir W. H. White,[1] "has varied from 40 to 50 per cent increase above the maximum power obtained by natural draft." The obvious conclusion is that a smaller boiler will, under forced draft, do the same work. In fact, forced draft has been generally applied, particularly in the marine service, "with a view to lessening the dimensions of the boiler. This is rendered possible by the increased temperature of the fire being able to transmit more heat per unit of surface, and to render the boiler, surface for surface, more efficient than it would be if a lower fire temperature were used. It is evident that if the boiler surfaces are more efficient, less heat will be wasted up the chimney."[1]

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[1] On the Speed Trials of Recent Warships. Sir W. H. White. Transactions of Institution of Naval Architects. 1886.

The horse-power of various types of boilers, based upon the area of heating surface shown by experiment to be necessary for the evaporation of 34.5 pounds of water from and at 212° per hour, is usually accepted as the nominal rating; owing to slight differences even between boilers of the same type, there is some latitude in this basis of measurement. The range of proportions generally adopted in practice is presented in Table No. 65.

Table No. 65. — Approximate Relation of Horse-Power and Heating Surface in Different Types of Boilers.

TYPE OF BOILER.	Square Feet of Heating Surface per H. P.	Coal per Square Foot Heating Surface per Hour	Relative Economy.	Relative Rapidity of Steaming.	Authority.
Water-tube,	10 to 12	0.3	1.00	1.00	Isherwood.
Tubular,	14 to 18	0.25	0.91	0.50	Isherwood.
Flue,	8 to 12	0.4	0.79	0.25	Prof. Trowbridge.
Plain cylinder,	6 to 10	0.5	0.69	0.20	
Locomotive,	12 to 16	0.275	0.85	0.55	
Vertical tubular,	15 to 20	0.25	0.80	0.60	

Radiation and Convection of Furnace Heat. — The ideal temperature of combustion of a coal consisting of 80 per cent carbon has already been shown by Table No. 20 to be 4,718° when the chemically necessary amount of air is supplied. If this supply be doubled, the temperature is reduced to 2,600°; in each case the temperature being measured above that of the air supplied to the furnace. These ideal temperatures pertain to the heart of the fire and can only exist in the furnace chamber if the fire is properly enclosed with radiating surface.

The ordinary boiler plate, with the hot gases on one side and water on the other, presents a very different condition, for it becomes a greedy absorber of heat, both radiant and convected. The air and gases, being poor conductors of heat, and absorbing but a very slight amount of the radiant heat, have only the power to increase the temperature of the surface with which they come in contact by the process of convection or carrying, which may be defined as "a transfer and diffusion of the state of heat in a fluid mass by means of the motion of the particles of that mass."[1]

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[1] On the Progress and Development of Marine Engineering. Wm. Parker. Transactions of Institution of Naval Architects. 1887.

The heat that is radiated from the fire is but feebly reciprocated from the plate surfaces of the boiler, since the plate is maintained at a temperature not much higher than that of the water inside. Under such conditions the heat which is radiated from the fuel upon the grate, together with that which is communicated by convection from the heated gases, is rapidly absorbed and carried off. It is, therefore, impossible to maintain in the furnace a temperature even near the maximum temperature of combustion. The radiation from the fuel, taking place, as it does, in straight lines, is thereby restricted in its effect to the grates, walls of the furnace chamber and the exposed portion of the boiler. The heat not thus lost is carried along by the products of combustion.

The exact proportional relation between the radiant and convected heat is difficult of determination, but Peclét assumed them to be equal as they leave the upper surface of the fuel. Upon this assumption and the formula derived by Dulong and Petit, the relation of radiant and convected heat has been calculated by D. K. Clark[2] for different rates of combustion. The conditions imposed are: complete combustion, no excess of air and a coal having a heat value of 14,700 B. T. U. The results are given in Table No. 66, with the average relative proportions existing between the two means of dissipation of heat.

Table No. 66. — Temperature and Heat of a Coal Fire in a State of Incandescence.

Coal Consumed per Square Foot of Grate per Hour. Pounds.	Temperature of Surface of Boiler Plate. Degrees Fahr.	Temperature of Surface of Fire. Degrees Fahr.	Approximately Calculated Distribution of the Heat of Combustion.				
			By Radiation.		By Convection of Gases.		Sum of Radiation and Convection. B. T. U.
			B. T. U.	Per Cent of Sum.	B. T. U.	Per Cent of Sum.	
5	350°	1,400°	53,960	74	19,160	26	73,120
10	350	1,550	102,500	70	43,510	30	146,010
20	350	1,705	198,400	67	96,080	33	294,480
40	350	1,857	378,650	64	209,850	36	588,500
80	350	2,009	721,800	61	455,400	39	1,177,200
120	350	2,097	1,049,000	59	714,050	41	1,763,050

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[1] A Manual of the Steam Engine. W. J. M. Rankine. London, 1885.

[2] The Steam Engine. D. K. Clark. London, &c., 1890.

Distribution of the Heat of Combustion. — It being the function of a boiler to utilize as much as possible of the heat generated by the combustion of the fuel in the furnace provided for the purpose, it is to be expected that the gases will be gradually cooled as they approach the chimney. The greater the decrease, and, other things equal, the lower the final temperature, the higher the efficiency. The temperature of the gases at different points in their passage across the heating surface of the boiler obviously depends upon the character, extent and temperature of that surface, and the initial temperature and velocity of the gases.

The absorption of practically all of the radiant and part of the convected heat by the surfaces exposed to the fire immediately lowers the temperature of the gases so that as they pass onward to the other portions of the heating surface their heating power is much decreased. The existence of this condition is rendered evident by the results of tests reported by Mr. Paul Havrez.[1] These tests were conducted upon a special boiler of the locomotive type, the barrel or cylindrical portion of which was divided into four sections of equal length, and so arranged that the evaporation in each could be ascertained separately. The general proportions of this boiler, together with the results, are presented in Table No. 67, from which it is evident that the surface of the firebox was about three times as efficient as that of the first section, and increasingly more for the other sections respectively.

Table No. 67. — Evaporation in Different Sections of Experimental Boiler.

Portion of Boiler.	Square Feet of Heating Surface.	Pounds of Water Evaporated per Hour per Square Foot of Heating Surface.	
		With Coke for Fuel.	With Briquettes for Fuel.
Firebox,	76.43	24.5	36.9
First section,	179	8.72	11.44
Second section,	179	4.42	5.72
Third section,	179	2.52	3.52
Fourth section,	179	1.68	2.31

The variation in the rate of absorption of heat from the gases, which takes place in their passage through the tubes of a marine boiler, is clearly shown by the results of tests conducted under the supervision of Mr. A. J. Durston, Engineer in Chief of the British Navy, as presented in Table No. 68. These results cover eight sets of records of temperatures taken by a Le Chatelier thermo-electric pyrometer inserted to various distances in the tubes. The boiler was being worked at its normal capacity, the rate of consumption of coal being about 17 pounds per square foot of grate. The temperature existing in the combustion chamber was 1,644°, and that just inside the tube 1,550°.

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[1] Proceedings of Institute of Civil Engineers. London, 1875.

Table No. 68. — Temperatures in Tubes of Marine Boiler.

LOCATION.	Temperature.	LOCATION.	Temperature.
1 inch from combustion chamber,	1,466°	1 ft. 2 in. from combustion chamber,	1,368°
2 inches from combustion chamber,	1,426	1 ft. 8 in. from combustion chamber,	1,295
3 inches from combustion chamber,	1,405	2 ft. 8 in. from combustion chamber,	1,198
4 inches from combustion chamber,	1,412	3 ft. 8 in. from combustion chamber,	1,106
5 inches from combustion chamber,	1,398	4 ft. 8 in. from combustion chamber,	1,015
6 inches from combustion chamber,	1,406	5 ft. 8 in. from combustion chamber,	926
7 inches from combustion chamber,	1,400	6 ft. 8 in. from combustion chamber,	887
8 inches from combustion chamber,	1,410	In smoke box,	782

The temperatures existing under usual conditions, in connection with an ordinary horizontal return tubular boiler, were very carefully ascertained by Mr. J. C. Hoadley,[1] and are presented in Table No. 69. The coal burned consisted of 82 per cent carbon completely consumed to carbonic acid, the steam pressure was about 45 pounds above the atmosphere and the air supply per pound of fuel was 21.28 pounds. The actual temperature shown to exist, as compared with that which the coal should be ideally capable of producing, with no excess of air, as previously indicated, is particularly to be noted.

Table No. 69. — Temperatures in Connection with Horizontal Return Tubular Boiler.

Location at which Temperature was Taken.	Temperature.
In heart of fire	2,426°
At bridge wall	1,341
In smoke box	368
Air admitted to furnace	78
Steam and water in boiler	292
Gases escaping to chimney	368

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

Disposition of Heat in Steam Boilers. — The theoretical heat losses incident to the combustion of a given amount of coal have already been considered in the preceding chapter. These indicate clearly the disposition of all the heat generated, with the exception of that lost through the brickwork. This loss is evidently variable and uncertain, but under ordinary conditions is between 4 and 20 per cent. Mr. J. C. Hoadley[1] conducted a series of experiments upon the setting of a horizontal return tubular boiler, by inserting thermometers to different depths in holes in the walls at 5 feet above the floor, — opposite to the body of the boiler and midway between the bridge wall and the pier. The maximum and minimum temperatures found between 8.30 a.m. and 4.30 p.m., at different depths, upon one day of the test, are presented in Table No. 70. Of course the temperature at the outer surface of the setting would be considerably less than that 4 inches inward, owing to the rapid radiation at the surface.

Table No. 70. — Temperatures of Brick Setting of Horizontal Return Tubular Boiler.

Place of Observation.	Minimum Temperature.	Maximum Temperature.
4 inches from outside of wall	140°	182°
16 inches from outside of wall	285	353
28 inches from outside of wall	297	460

The disposition of the heat generated in the furnace of a steam boiler, as determined by several investigators for different types of boilers, is indicated in Table No. 71. The boiler experimented on by Dr. Bunté was of ordinary character; that designated A and reported upon by MM. Scheurer and Meunier was of the French type, having three heaters with six feed-heater tubes at one side; while the results given under B are the average of experiments upon four different types of boilers. Boiler C, experimented upon by Messrs. Donkin and Kennedy, was of vertical tubular construction with an internal firebox; boiler D was of the locomotive pattern; while E was of "elephant" type, provided with Green's economizer. The boiler tested by Hoadley was of the ordinary multitubular form, provided with a special air-heating arrangement which lowered the temperature of the flue gases about 213°, and raised that of the air supplied to the furnace about 300°. These abstractors account for the high efficiency secured, which might have been closely approached by boiler E had it not been for the excessive loss through the brickwork.

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

All figures are given in percentages of the total amount of heat accounted for, The percentage of heat, which is indicated to have been disposed of in heating and evaporating the water, is a direct measure of the efficiency of the boiler.

Table No. 71. — Disposition of Heat in Steam Boilers.

DISPOSITION OF HEAT.	AUTHORITY.						
	Bunté.	Scheurer and Meunier.		Donkin and Kennedy.			Hoadley.
		A	B	C	D	E	
Waste in flue gases including evaporation of moisture in coal and heating vapor in air when these losses are not separately given,	18.6	5.5	14.8	9.4	22.5	6.5	5.04
Evaporating moisture in coal,	3.5	2.5	6.1	0.1	0.1	0.0	1.55
Heating vapor in air,							0.18
Imperfect combustion,	8.0	6.0		12.7	0.0	0.0	1.44
Clinker and ash,	4.1			0.1	0.2	0.0	
Radiation and heat not otherwise accounted for,	7.6	23.5	13.4	13.9	11.0	15.0	4.00
Heating and evaporation of water,	58.2	61.0	65.7	63.8	66.2	78.5	87.79

Sources of Efficiency. — With a given fuel, and otherwise identical conditions, the efficiency of a boiler is most largely dependent upon the relation of its grate area to its heating surface, and upon the rate of combustion of the fuel. Under the same conditions of boiler and fuel, the greater the quantity consumed per hour the greater is the amount of water evaporated per hour. But at the same time the quantity of water evaporated per pound of fuel decreases because of the higher temperature of the escaping gases. This loss can only be diminished by increasing the heating surface either in the boiler or in a separate heater, while the decreased draft, due to lowered temperature, is easily made good by mechanical means. It is to be carefully noted that all this relates to a condition of increased boiler capacity, resulting from a greater coal consumption. But when the capacity and coal consumption are maintained practically constant, a higher efficiency per pound of fuel may actually be obtained by a moderate reduction of grate area, whereby the surface ratio is increased, and a corresponding increase in the rate of combustion per square foot of grate. This matter of the relation between the surface ratio and the rate of combustion and its economical influence, as indicated in the evaporation per unit of fuel, will be discussed at considerable length in the succeeding chapter.

The quality of the fuel must to a certain extent enter into the problem of boiler design to secure the highest efficiency. The desirable features to give the best results with low-grade fuels are concisely stated by Mr. W. H. Bryan[1] to be —

"A. Ample draft; 1 inch of water or even more. Good results cannot be secured with drafts less than one-half inch. Good draft and thick beds of fuel permit high firebox temperatures, which we have found absolutely necessary.

"B. Large ratio of heating to grate surface, so that while burning coal at a high rate per square foot of grate per hour, there is sufficient heating surface to reduce the temperature of the flue gases to 450° Fahr. or less.

"C. The combustion chamber should, if possible, be separate from the heating surfaces, so as to avoid their cooling effect. It should be quite deep — 30 inches or more."

As indicating the importance of the draft, and the practical necessity of mechanical means for creating the same, Mr. Bryan states: "To secure the very highest results, the gases, after leaving the boiler-heating surfaces at not exceeding 500°, should be passed through feed-water economizers and thence through air heaters. The feed-water, leaving the ordinary exhaust heater at a little above 200° Fahr., may be raised to over 300° in the economizer, and the heated gases reduced to 250°, or less. This reduction in temperature, of course, destroys the usefulness of these gases as draft producers, unless the chimney is very tall. The draft, however, can be better produced by exhaust fans, which draw the air through and out of the furnace and economizer, and discharge the gases at such a height above the roof that they will not be objectionable; thus doing away entirely with the necessity for high chimneys. Still better economy may be secured by placing air heaters in the smoke flue, beyond the fan or between it and the economizer. Through this the air, entering the ashpit for purposes of combustion, may be drawn, so that the heated gases are finally discharged at a temperature but little above that of the atmosphere. The speed of the fan may be controlled by an automatic regulator, which increases the speed of the fan engine as the steam pressure drops, and reduces it as the pressure increases; thus performing all the functions of an automatic damper regulator. This plan is not experimental or untried, but has already been adopted in numerous large plants."

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[1] Boiler Efficiency, Capacity and Smokelessness, with Low-Grade Fuels. Wm. H. Bryan. A paper read before the Engineers' Club of St. Louis, Oct. 21, 1896.

The influence of the relation between area of grate and of heating surface was very carefully investigated by Mr. D. K. Clark,[1] in the case of a locomotive boiler using coke. From these tests he deduced,—

"*First.* That, assuming throughout a constant efficiency of the fuel or proportion of water evaporated to the fuel, the evaporative performance of a locomotive boiler, or the quantity of water which it is capable of evaporating per hour, *decreases* directly as the grate area is increased; that is to say, the larger the grate the smaller is the evaporation of water when the efficiency of the fuel is the same, even with the same heating surface.

"*Second.* That the evaporative performance *increases* directly as the square of the heating surface, with the same area of grate and efficiency of fuel.

"*Third.* The necessary heating surface *increases* directly as the square root of the performance; that is to say, for example, for four times the performance, with the same efficiency, twice the heating surface only is required.

"*Fourth.* The necessary heating surface *increases* directly as the square root of the grate, with the same efficiency; that is to say, for instance, if the grate be enlarged to four times its first area, twice the heating surface would be required, and would be sufficient to evaporate the same quantity of water per hour with the same efficiency of fuel."

The relation between the area of grate and of heating surface, which has already been expressed as the "surface ratio," may be thus represented: —

$$\text{Surface ratio} = \frac{\text{Area of heating surface.}}{\text{Area of grate surface.}}$$

This ratio naturally varies according to the type of boiler, the general practice being about as indicated in Table No. 72.

Table No. 72. — Surface Ratios of Steam Boilers.

TYPE OF BOILER.	Area of Heating Surface when Grate Area = 1.
Marine return tubular	25 to 38
Lancashire	26 to 33
Cornish	25 to 40
Modified locomotive type	30 to 34
Horizontal return tubular	30 to 50
Water-tube	35 to 65
Horizontal internally fired multi-tubular	25 to 45
Locomotive	60 to 90
Plain cylinder	10 to 15

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[1] The Steam Engine. D. K. Clark. London, &c., 1890.

As it has already been shown,— that when the total coal consumption is increased the heating surface must also be increased to maintain the same efficiency,— so the converse must be evident: that with a given rate of combustion it is not economical to increase the area of heating surface beyond certain limits. These limits must of necessity be determined by experiment.

Mr. D. K. Clark[1] summarized his deductions from a large number of tests of boilers of different types in the following formulæ: —

Stationary boilers . . .	$w = 0.0222\, r^2 + 9.56\, c.$
Marine boilers . . .	$w = 0.016\, r^2 + 10.25\, c.$
Portable engine boilers . .	$w = 0.008\, r^2 + 8.6\, c.$
Locomotive boilers (coal-burning),	$w = 0.009\, r^2 + 9.7\, c.$
Locomotive boilers (coke-burning),	$w = 0.0178\, r^2 + 7.94\, c.$

In which w = weight of water in pounds per square foot of grate per hour.
c = pounds of fuel per square foot of grate per hour.
r = ratio of heating to grate surface.

The water is taken as evaporated from and at 212°.

The ratio of grate surface to heating surface being one of the factors, it is evident that by means of these formulæ the evaporation per square foot of heating surface may also be obtained. There are minimum rates of consumption of fuel below which these formulæ are not applicable. The limit varies for each kind of boiler and with the surface ratio. It is imposed by the fact that the maximum evaporative power of fuel is a fixed quantity, and is naturally at that point where the reduction of the rate of combustion for a given ratio procures the absorption into the boiler of the whole of the proportion of the heat which is available for evaporation. In the combustion of good coal the limit of evaporative efficiency may be taken as measured by 12.5 pounds of water from and at 212°. Table No. 73, based upon these formulæ, presents the effect of increasing rates of combustion with different surface ratios. While for a given boiler and surface ratio this table indicates that the evaporative efficiency decreases as the rate of combustion is increased, it is to be noted that the capacity of the boiler is increased also, and that by a proper application of more heating surface the efficiency may be maintained. In other words, a boiler and its appurtenances should be designed for a given rate of combustion. A high rate of combustion is not, therefore, an indication of low efficiency; but, as will be shown later, is, with a proper surface ratio, one of the important factors in attaining economy in boiler practice.

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[1] The Steam Engine. D. K. Clark. London, &c., 1890.

Table No. 73.—Evaporative Performance of Steam Boilers for Increasing Rates of Combustion and Different Surface Ratios and Best Coal and Coke.

	KIND OF BOILER.	Water from and at 212° per Hour.	Fuel per Square Foot of Grate per Hour in Pounds.						
			5	10	15	20	30	40	50
For Best Coal. Surface Ratio = 30.	Stationary,	Per square foot of grate,	62.5*	116	163	211	307	402	498
	Stationary,	Per pound of coal,	12.5	11.56	10.89	10.56	10.23	10.06	9.96
	Marine,	Per square foot of grate,	62.5*	117	168	219	322	424	527
	Marine,	Per pound of coal,	12.5	11.69	11.25	10.95	10.69	10.61	10.54
	Portable,	Per square foot of grate,	50.0	93	136	179	265	351	437
	Portable,	Per pound of coal,	10.0	9.3	9.01	8.95	8.83	8.77	8.74
	Locomotive,	Per square foot of grate,	57.0	105	154	202	299	396	493
	Locomotive,	Per pound of coke,	11.4	10.5	10.26	10.10	9.97	9.90	9.86
Surface Ratio = 50.	Stationary,	Per square foot of grate,	62.5*	125*	187.5*	247	342	438	534
	Stationary,	Per pound of coal,	12.5	12.5	12.5	12.33	11.41	10.95	10.67
	Marine,	Per square foot of grate,	62.5*	125*	187.5*	245	348	450	552
	Marine,	Per pound of coal,	12.5	12.5	12.5	12.25	11.58	11.25	11.05
	Portable,	Per square foot of grate,	62.5*	106	149	192	278	364	450
	Portable,	Per pound of coal,	12.5	10.6	9.93	9.6	9.27	9.10	9.00
	Locomotive,	Per square foot of grate,	62.5*	120	168	217	314	411	508
	Locomotive,	Per pound of coke,	12.5	11.95	11.20	10.85	10.45	10.26	10.15

* These quantities fall below the scope of the formulæ for the water, as explained in the text.

By the same process of reasoning employed in the previous chapter, in the discussion of the influence of the air supply, it is evident that the higher temperature of the escaping gases resulting from an increased total coal consumption is due to the increased supply of air and the higher velocity which, therefore, ensues. Although this higher temperature is but the result of more rapid combustion, it is at the same time an absolute necessity when a chimney alone is depended upon to create the draft. This is because the draft required for the increased combustion and air supply can only be secured, in the case of a chimney, by raising the temperature of the chimney gases. Under these conditions any attempt to reduce this final temperature by the addition of heating surface must of necessity tend to reduce the rate of combustion. Furthermore, the influence upon the draft of such additional surface will be twofold: it will be reduced both because of the lower temperature in the chimney, and because of the increased resistance due to the extended surface. The rate of combustion can only be maintained by supplementing the draft, which may be done by introducing a fan either for forcing in the air or withdrawing the gases.

But, with sufficient draft, higher efficiency may evidently be secured by increasing the heating surface, with the limitation that the surface shall not be so great as to cool the gases too near to the temperature of the steam; for it is probable that there can be no active transmission of heat from the gases without to the water within a boiler, with less than 75° difference of temperature. Nevertheless, one of the vital principles underlying the attainment of economy in the generation of steam is a low temperature of the escaping gases.

The opportunity for securing more economic results by reducing the temperature of the flue gases is well evidenced in the results of seventeen independent boiler tests, by Messrs. Donkin and Kennedy. They found the heat lost up the stack, where no economizer was used, to range between 9.4 per cent and 31.8 per cent of the total heat of combustion, the average being 20.3 per cent. It is, therefore, evident that in this direction lies one of the greatest opportunities for increasing boiler efficiency. Although additional surface may be obtained by reducing the size of the tubes and increasing their number, or by ribbing them, or introducing retarders, it is usually customary to abstract the surplus heat from the gases by some means in a sense independent of the boiler. It may then take the form of a feed-water heater, otherwise known as an economizer, or the form of a device for abstracting the heat from the gases and transferring it to the air supplied to the fuel, or both. The results obtained by either of these methods have, in the case of chimney draft, always been restricted by the cost of the excessively high chimney necessary to produce the requisite draft with the decreased temperature and increased resistance. The simplicity and efficiency of mechanical draft, however, obviates this difficulty, and makes possible the attainment of much lower final temperatures of the flue gases with a corresponding increase of efficiency.

Flue Feed-Water Heaters or Economizers. — The modern type of fuel economizer consists of a series of tubes, made up in sections, connected at the ends and placed in a brick chamber, through which the gases pass from the boiler to the chimney or fan. Feed-water is forced through the tubes, while the gases circulate around them. The difference between the economizers of different makes lies principally in the proportions, the design of end connections, and the position of the tubes. As is evident from Table No. 76, and the accompanying explanation, no economizer is complete without some device for continuously or periodically removing the soot from the exterior of the tubes.

The economy resulting from the introduction of an economizer when the draft is sufficient naturally depends upon the normal temperature of the flue gases escaping from the boilers, and of the feed-water supplied to the economizer. The percentage of gain may be determined by the following formula:—

$$\text{Gain, in per cent,} = \frac{100\,(T - t)}{H - t}$$

In which T = heat units in one pound of feed-water above 0° after heating;
t = heat units in one pound of feed-water above 0° before heating;
H = heat units in one pound of steam of boiler pressure above 0°.

Table No. 74, calculated by this formula, indicates the saving under different conditions of feed-water, with steam of 70 pounds boiler pressure. Of course the greatest economy will appear where the temperature of the flue gases was originally the highest and that of the feed-water the lowest. Even with a low temperature of the flue gases, an economizer will usually show results that warrant its introduction.

Table No. 74.—Percentage of Saving in Fuel by Heating Feed-Water. Steam at 70 Pounds Gauge Pressure.

Initial Temperature of Feed-Water.	TEMPERATURE TO WHICH FEED-WATER IS HEATED.														
	100°	110°	120°	130°	140°	150°	160°	170°	180°	190°	200°	210°	220°	250°	300°
35°	5.53	6.38	7.24	8.09	8.95	9.89	10.66	11.52	12.38	13.24	14.09	14.95	15.81	19.40	29.34
40	5.12	5.97	6.84	7.69	8.56	9.42	10.28	11.14	12.00	12.87	13.73	14.59	15.45	18.89	28.78
45	4.71	5.57	6.44	7.30	8.16	9.03	9.90	10.76	11.62	12.49	13.36	14.22	15.09	18.37	28.22
50	4.30	5.16	6.03	6.89	7.76	8.64	9.51	10.38	11.24	12.11	12.98	13.85	14.72	17.87	27.67
55	3.89	4.75	5.63	6.49	7.37	8.24	9.11	9.99	10.85	11.73	12.60	13.48	14.35	17.38	27.12
60	3.47	4.34	5.21	6.08	6.96	7.84	8.72	9.60	10.47	11.34	12.22	13.10	13.98	16.86	26.56
65	3.05	3.92	4.80	5.67	6.56	7.44	8.32	9.20	10.08	10.96	11.84	12.72	13.60	16.35	26.02
70	2.62	3.50	4.38	5.26	6.15	7.03	7.92	8.80	9.68	10.57	11.45	12.34	13.22	15.84	25.47
75	2.19	3.07	3.96	4.84	5.73	6.62	7.51	8.40	9.28	10.17	11.06	11.95	12.84	15.33	24.92
80	1.76	2.65	3.54	4.42	5.32	6.21	7.11	8.00	8.88	9.78	10.67	11.57	12.46	14.82	24.37
85	1.30	2.22	3.11	4.00	4.90	5.80	6.70	7.59	8.48	9.38	10.28	11.18	12.07	14.32	23.82
90	0.89	1.78	2.68	3.58	4.48	5.38	6.28	7.18	8.07	8.98	9.88	10.78	11.68	13.81	23.27
95	0.45	1.34	2.25	3.15	4.05	4.96	5.86	6.77	7.66	8.57	9.47	10.38	11.29	13.31	22.73
100	0.00	0.90	1.81	2.71	3.62	4.53	5.44	6.35	7.25	8.16	9.07	9.98	10.88	12.80	22.18

In Table No. 75 are presented the results of a number of tests by George H. Barrus[1] of boiler plants, of which an economizer in each case formed a part. In the first four cases, designated A, B, C and D, the temperature of the gases as they leave the boiler is comparatively low, — namely, 394°, — but the initial temperature of the feed-water is raised 92° on the average, and the evaporation increased 9.9 per cent per pound of coal. If this result is applied to a 1,000-horse-power plant, and the cost of Cumberland coal for a day's run of 10 hours taken as $63.38, per Table No. 58, the daily saving would be $6.27 per day. For a year of 308 working days this represents a total saving of $1,931.16. An economizer equipment sufficient to secure the above result would probably cost from $7,000 to $8,000. The saving would represent an annual return of 24 to 28 per cent, certainly sufficient to warrant careful consideration.

Table No. 75. — Tests with Economizers.

	Designation of Boiler.				
	A	B	C	D	E
Area of heating surface, boiler, square feet,	1,894	4,058	5,592	3,126	1,880
Area of heating surface, economizer, square feet,	1,600	1,920	1,280	1,600	1,600
Temperature of gases leaving boiler, degrees,	376	361	403	435	618
Temperature of gases leaving economizer, degrees,	231	254	299	279	365
Temperature of feed-water entering economizer, degrees,	95	79	111	84	88
Temperature of feed-water entering boiler, degrees,	175	145	169	196	225
Increased evaporation produced by economizer, per cent,	10.5	7.0	9.3	12.8	29.0

Upon the same basis of calculation, the case designated E would show an annual return of about 75 per cent of the investment. Although this result is only approximate, it is sufficiently near the truth to indicate the indisputable economic advantage of the economizer. The cost of the means for producing the draft requisite to such results will be found to be less in the case of a fan than of a chimney, while the former will in addition possess advantages which make it much the more desirable of the two.

At the steam plant of the Cheney Brothers silk mills at South Manchester, Conn., which is provided with an economizer and Sturtevant Mechanical Draft apparatus, 45 cubic feet of water is heated per minute and used in the boilers, while an additional 50 cubic feet is heated and utilized in the dyehouse. The whole quantity is raised from an initial temperature of 112° to 211°. The heating of the feed-water alone is sufficient to cause a saving in the evaporative work of the boiler amounting to 10 per cent. So that the total saving, including the heat utilized in heating water for the dyehouse, is over 20 per cent. In a plant of 600 horse-power running 10 hours per day, and using coal at $4.00 per ton, this represents an annual saving of about $2,000 per year, an excellent return on the additional investment required to install the economizer plant.

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[1] Boiler Tests. George H. Barrus. Boston, 1891.

The influence of soot, and the necessity of frequent if not continuous cleaning of the surfaces of an economizer, was clearly shown by M. W. Grosseteste[1] in a three weeks' test with smoky coal upon a Green economizer, consisting of a series of vertical pipes arranged to be cleaned externally by automatic scrapers. The apparatus had been at work for seven weeks continuously without having been cleaned, and had accumulated a half-inch coating of soot and ash. It was observed in this condition throughout the first week. During the second week it was cleaned twice every day, but during the third week, after having been cleaned on Monday morning, it was worked continuously without further cleaning. The results presented in Table No. 76 show the necessity of cleaning.

Table No. 76. — Influence of the State of the Surface of an Economizer.

TIME. (February and March.)		Temperature of Feed-Water.			Temperature of Gaseous Products.			Coal Consumed per Hour. Pounds.	Water Evaporated from 32° per Hour. Pounds.	Water per Pound of Coal. Pounds.
		Entering Economizer.	Leaving Economizer.	Difference.	Entering Economizer.	Leaving Economizer.	Difference.			
First week,		73.5°	161.5°	88.0°	849°	261°	588°	214	1,424	6.65
Second week,		77.0	230.0	153.0	882	297	585	216	1,525	7.06
Third week.	Monday,	73.4	196.0	122.6	831	284	547	213	1,428	6.70
	Tuesday,	73.4	181.4	108.0	871	309	562			
	Wednesday,	79.0	178.0	99.0	—	—	—			
	Thursday,	80.6	170.6	90.0	952	329	623			
	Friday,	80.6	169.0	88.4	889	338	551			
	Saturday,	79.0	172.4	93.4	901	351	550			

Note to Table. — The averages for the first and second weeks are exclusive of Monday.

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[1] Bulletin de la Société Industrielle de Mulhouse, Vol. XXXIX. 1869.

Air Heaters or Abstractors. — The air heater, heat abstractor, warm-blast apparatus or hot-draft apparatus, as it is variously called, generally consists of some arrangement of pipes, through or across which the hot gases pass direct from the uptake, heating thereby the air supply for the furnace, which passes respectively across or through the pipes; the object being to abstract from the gases as much heat as is practical and transfer it to the air before it enters the furnace, thereby securing a higher temperature and increased evaporative efficiency.

Such an apparatus is virtually the equivalent in results of an economizer, and is the only practical means of reducing the waste of heat in the flue gases when large quantities of warm water are not in demand, as must be the case if an economizer is to show efficient results. The ideal arrangement, however, consists of a combination of economizer and abstractor, whereby the air supply to the furnace may be heated as well as the feed-water for the boiler, and all the heat practicable thus abstracted from the flue gases. For such results chimney draft is ordinarily inadequate and mechanical means must be resorted to, to overcome the increased resistance.

Doubtless the most comprehensive test ever conducted upon an apparatus of this character was that undertaken by Mr. J. C. Hoadley,[1] in the interest of a number of mill owners, and intended to determine the efficiency of the Marland apparatus for heating the air supply. The original apparatus was applied to a single horizontal return tubular boiler, 60 inches in diameter, with 65 tubes 3½ inches in outside diameter, and 20 feet long. This boiler was of the regular type in use in the Pacific Mills, Lawrence, Mass., where the tests were made, except that upon its top were placed two abstractors, one upon either side, 3 feet apart, extending the entire length of the boiler. Each abstractor contained 120 lap-welded tubes, 2 inches in outside diameter and 20 feet long, enclosed in a brick setting. Surrounding each pipe was a 3-inch tube of thin iron. By proper arrangement of the heads into which the 2-inch pipes were expanded, in connection with the uptake, a passage was provided for the flue gases through these pipes and thence to the blower which produced the requisite draft. The 3-inch tubes were shorter than the 2-inch, and were arranged at their ends so that air for the furnace could pass to them and thence through the annular space between the two tubes, becoming heated by the gases in the inner tube. This apparatus was known as Warm-Blast No. 1. Alongside this boiler, and operating under the same conditions, was a boiler of the regular type, designated in the report as Pacific Boiler.

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

Extended experiment having shown Warm Blast No. 1 to be incapable of reducing the temperature of the escaping gases below 160°, the Pacific Boiler was, accordingly, converted into Warm-Blast No. 2 by placing upon its top two abstractors differing from the previous ones, and constructed substantially as follows: 2-inch spiral-locked tubes were provided for the passage of the hot gases, while the 3-inch tubes were replaced by a series of deflectors set at right angles to the tubes, which passed through them. These deflectors were so arranged that air entering at the top must descend across and among the 2-inch tubes, which had 1-inch spaces between them, pass under the first deflector, then rise in the same manner and pass over the second deflector, and so on, until the air passed to the ashpit.

The comparative temperatures found to exist in connection with the Pacific and Warm-Blast No. 1 boilers, properly reduced for comparison, are presented in Table No. 77. These figures alone seem to point to the efficiency of the warm-blast apparatus; but Table No. 78, comprising the important economic results, serves to indicate more definitely the relative efficiency of the various arrangements, and to prove the marked advantage of the warm-blast arrangement. Careful tests showed that the power consumed in driving the blower was about 1 per cent of the whole power produced by the boiler in combination with a good steam engine. This should be compared with the much larger expenditure required to produce the draft by means of a chimney.

Table No. 77. — Comparative Temperatures in Pacific Boiler and Warm-Blast Boiler No. 1.

Location at which Temperature was Taken.	Temperatures.		
	Pacific Boiler.	Warm-Blast Boiler.	Difference.
In heart of fire	2,493°	2,793°	300
At bridge wall	1,340	1,600	260
At pier	895	1,050	155
In smoke-box	373	375	2
Air admitted to furnace	32	332	300
Steam and water in boiler	300	300	0
Gases escaping to chimney	373	162	211
External air	32	32	0
Gases cooled, Warm-Blast Boiler			213
Air warmed, Warm-Blast Boiler			300

The results obtained by the use of other forms of air heaters, such as the Howden and the Ellis & Eaves, when used in connection with a fan for producing draft, will be discussed in a succeeding chapter.

Table No. 78.—Results of Tests with Pacific Boilers and Warm-Blast Boilers.

		With Anthracite Coal.			With Bituminous Coal.	
		Pacific Boiler.	Warm-Blast No. 1.	Warm-Blast No. 2.	Pacific Boiler.	Warm-Blast No. 1.
Mean temperature of external air, days,	degrees,	78.3°	34°	49°	71°	34.2°
Temperature of smoke-box,	degrees,	368.3	396.9	377	376.9	397.4
Temperature of escaping gases,	degrees,	368.3	189	164	376.9	196
Gases cooled by abstractors,	degrees,	0	207.9	213	0	201.4
Air warmed by abstractors,	degrees,	0	303.7	285	0	315.5
Temperature of air supplied to furnace,	degrees,	78.3	337.7	334	71	349.5
Temperature of steam,	degrees,	297.5	361.1	291.2	297.3	322.6
Loss at chimney,	per cent,	17.75	15.00	12.83	17.03	14.24
Loss by radiation from brickwork,	per cent,	2.64	4.00	4.00	3.39	4.00
Loss by imperfect combustion,	per cent,	2.13	0.63	1.43	2.85	1.06
Total loss by above three causes,	per cent,	22.52	19.63	18.26	23.27	19.30
Pounds of flue gases per pound of coal,		22.39	23.49	24.17	25.23	28.37
Efficiency, reduced to common basis,	per cent,	68.87	78.18	81.43	64.61	77.59
Difference of efficiency, points gained by Warm-Blast over Pacific Boiler,			9.31	12.56		12.98
Ratio of gain to the larger quantity $\left(\frac{9.31}{78.18}=11.9\%\right)$			11.9	15.4		16.7
Ratio of gain to the smaller quantity $\left(\frac{9.31}{68.87}=13.5\%\right)$			13.5	18.2		20.1

Increased Tube Heating Efficiency.—With fire tubes of a given length the amount of heat transmitted to the water within the boiler must be dependent upon the temperature and velocity of the gases, the amount of surface exposed, and the completeness with which they are forced into contact with it. In other words, with the same velocity and temperature, a given length of tube will be efficient in proportion as it presents absorbing surface for receiving the heat of the gases, and as those gases are compelled to come in contact with it. Two methods are in use for accomplishing this result. The first consists in fitting within a regular boiler tube a strip of thin sheet-iron, equal in width to the internal diameter of the tube, and twisted so as to form a helix of long pitch, making only two or three turns in the length of the tube. The effect of this arrangement—the strip being known as a "retarder"—is to break up the current of gas and cause all portions of its volume to touch the inner surface of the tube. At the same time the retarder itself is intensely heated, and rapidly radiates its heat through the tube to the water. The dual effect of the retarder is to materially increase the evaporative power and efficiency of the boiler.

The economic result of the use of retarders is shown by the tests of Mr. J. M. Whitham[1] upon a 100-horse-power horizontal tubular boiler operated at from about 50 per cent below to about 140 per cent above its rated capacity. The evident result is a reduction in the temperature of the flue gases, with a corresponding decrease in the coal consumption. This is shown in Table No. 79.

But, evidently, as the name "retarder" implies, this result cannot be attained without an increase in the draft. In this connection Mr. Whitham presents results, given in Table No. 80, showing the different drafts and resistances.

Table No. 79. — Reduction in Temperature of Flue Gases and in Coal Consumption by the Use of Retarders.

Horse-Power Developed.	Reduction in Temperature of Flue Gases. Degrees Fahr.	Reduction in Coal Consumption. Per cent.
52	20	0.0
75	53	0.0
100	32	3.2
125	46	4.0
150	19	3.3
170	59	3.6
200	36	4.1
225	26	8.6
239	123	18.4

Naturally, this additional draft can be most readily obtained when mechanical means are employed. In fact, retarders have been most extensively introduced where mechanical draft is in use.

Mr. Whitham's general deductions regarding retarders are that they interpose a resistance varying with the rate of combustion; that they reduce the temperature of the flue gases, and increase the effectiveness of the heating surface; that they should not be used where the draft is small; that they can be used to advantage in plants using a fan, and that they may show from 5 to 10 per cent advantage whenever the boiler plant is pushed and the draft is strong.

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[1] The Effect of Retarders in Fire Tubes of Steam Boilers. J. M. Whitham. Transactions American Society of Mechanical Engineers, Vol. XVII.

Table No. 80.—Draft and Resistance when Retarders are Used.

ITEMS.	Draft or Resistance in Inches of Water.
Furnace draft	0.30
Resistance of pass under boilers and through tubes without retarders,	0.27
Total draft of stack if no top is used	0.57
Resistance due to having retarders	0.31
Total draft if there is no return pass and retarders are used,	0.88
Increased resistance due to return pass over top of boilers,	0.07

The second method of increasing the efficiency of fire tubes is more direct, and consists in a special construction of the tube itself. Such is the case in the Serve tube, which is outwardly cylindrical, but from its inner or fire surface a number of equidistant radial ribs parallel to the axis converge toward the centre. The radial length of the ribs is usually about one-fifth of the external diameter of the tube, and they are seven or eight in number, according to the external diameter. The superior economy of these tubes is accounted for upon the theory that the ribs break up the column of gases, and by means of their extended surface extract heat from all parts of it. A six-day comparative test of the efficiency of plain and ribbed tubes, under practically identical conditions, was made by Mr. H. B. Roelker,[1] and the general results are presented in Table No. 81. In some cases retarders are used with these tubes, securing thereby even better contact of the air, because of its being forced to pass through the spaces between the retarder and the ribs.

Table No. 81.—Comparative Tests of Efficiency of Ribbed and Plain Fire Tubes.

DURATION OF TESTS.	Manner of Producing Draft.	Intensity of Draft. Inches Water.	Pounds of Water Evaporated per Pound of Coal.	
			Plain Tubes.	Ribbed Tubes.
Two 8-hour tests	Natural	⅛	5.08	7.35
Two 8-hour tests	Mechanical	½	5.98	7.60
Two 8-hour tests	Mechanical	⅞	4.68	6.75
One 8-hour test	Mechanical	1 3-16		7.41
One 8-hour test	Mechanical	1 9-16		6.52

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[1] Serve's Ribbed Boiler Tube. Passed Assistant Engineer G. S. Willits, U. S. Navy. Journal of American Society of Naval Engineers, August, 1891.

Of course the highest relative efficiency of such devices will be shown when the tubes are comparatively short and the gases with plain tubes are rejected at a high temperature. But under all ordinary conditions there can be no question of their efficiency. They are of especial advantage when space or the design of the boiler forbids the convenient introduction of additional heating surface in any other manner, as is usually the case in marine boiler practice. They cannot, however, be advantageously used when the draft is small, but are sources of decided economy in mechanical draft plants. In fact, under most conditions mechanical draft is necessary to their success, and with it they are quite extensively employed.

Mechanical Stokers. — The higher efficiency attained when the firing is in small amounts at frequent intervals, and when the fire is carefully maintained in the best possible conditions, points to the results which should ensue from the employment of a proper method of continuously feeding the fire by mechanical means. The mechanical stoker, as a substitute for hand-firing, possesses many advantages, but they can only be realized when the stoker is properly suited to its particular work and is intelligently operated. Its advantages may be summarized as, —

First. Adaptability to the economical combustion of the cheapest grades of fuel.

Second. Saving in labor of firing.

Third. Economy in combustion even with forced firing, under proper management.

Fourth. Constancy and uniformity of the furnace conditions.

Fifth. Smokelessness.

Three principal types of mechanical stokers are to be found in use.

The under feed, by which the fuel is, by screw or plunger, forced upward from beneath. In the common forms the fuel is thus supplied along the centre of the length of the grate, and as it is forced upward falls over to the sides and thus forms a long mound, thin at the sides of the grate and of considerable thickness in the centre. This thickness necessitates a very strong under-grate blast, which can only be secured by the use of a blower, and is generally applied in greatest volume at or near the centre.

The inclined over-feed type generally consists of a sloping grate, the highest portion being at the front of the boiler, where the coal is fed. The grate bars are usually so constructed and arranged that they may be periodically moved so as to feed the fuel along and down the surface. The motive power by which this practically continuous movement and feeding process is maintained is, in most cases, derived from a small independent engine, although hand power is sometimes used in small plants. Evidently this latter method of operating results in a much less frequent movement of the coal. The best results with this form of mechanical stoker are usually obtained when forced draft is used and the air is admitted either through the grate bars themselves, or between them from the space beneath. With either arrangement there is an excellent opportunity for the most perfect distribution of the air and its intimate contact with the fuel.

The third type of mechanical stoker consists of a chain grate upon which the fuel is fed at the front of the boiler, and which by its slow progress toward the bridge wall gives the fuel an excellent and undisturbed opportunity for complete combustion. It is particularly adapted for the lowest grades of fuel, and in its most perfect form is so arranged that the fuel, at different points in its progress, receives its air supply in different amounts and under different pressures, each best suited to the given stage of the combustion. The proper introduction and regulation of the air requires that it should be supplied by a blower, which thus forms an inherent part of such a plant. Comparisons between mechanical stoking and hand-firing, as well as between different forms of mechanical stokers, demand that the conditions shall be practically identical. In Table No. 82 are presented the results of several comparative tests made under such conditions, each with a different form of mechanical stoker. They serve to show the undeniable economy resulting from this method of feeding coal. Although several forms of mechanical stokers are included in the list, the tests are, for obvious reasons, designated only by distinguishing letters, each letter covering the results with both hand-firing and mechanical stoking under similar conditions.

Table No. 82.—Tests of Mechanical Stokers.

	A		B		C		D	
	Hand.	Stoker.	Hand.	Stoker.	Hand.	Stoker.	Hand.	Stoker.
Coal per hour, pounds,	2,022	1,422	2,022	1,800	428	432	857	771
Coal per hour per sq. ft. grate, pounds,	25.3	28.4	25.3	29.0			29.1	30.9
Water evaporated per hour per pound of coal from and at 212°, pounds,	7.11	8.70	7.11	9.12	7.68	8.80	8.81	10.29
Water evaporated per hour per pound of combustible from and at 212°, pounds,	7.77	9.67	7.77	10.07	8.96	10.01		

Constant attention is necessary in mechanical firing in order to regulate the rate of feed to the rate of evaporation; but the total amount of labor is far less than that required in hand-firing. When bituminous coal is mechanically fired there can be but little question that the plant, if of reasonable size, will operate with sufficient economy to pay a good return on the extra investment required for the stoking apparatus.

Powdered Fuel Furnaces. — Coal, in the form of dust, fed to the boiler furnace in a current of air, has to some extent been employed for the purposes of steam generation. The arrangement usually comprises a device for reducing the coal to an impalpable powder. It is then fed, together with air ordinarily supplied by a fan, into the front of the furnace. Theoretically, this method appears to have certain advantages which should make it successful. There is opportunity for instantaneous combustion, and the most intimate contact of the air, whereby the minimum amount may be employed. There should be no loss by decrepitation; but this is more than offset by the tendency to blow the dust in an unconsumed state directly up the chimney. The results indicate, however, that with such methods as have been tested the gain, if any, is more than counterbalanced by the added expense.

Influence of Mechanical Draft on the Ultimate Efficiency of Steam Boilers.— Several, although by no means all, of the advantages of mechanical draft as a means of increasing boiler efficiency and capacity have already been pointed out. By its use the draft is rendered positive, and the air required for a given weight of fuel may be reduced to a minimum. As stated by Mr. Richard Sennett,[1] it "tends to promote economy of fuel in consequence of the better supply of air and the higher temperature at which the fires are worked."

The efficiency of a boiler plant from a commercial standpoint — and that is the point from which it must ultimately be judged — concerns not alone the cost of the fuel, and the cost of handling and firing the same, but also the cost of the boiler plant itself, including the space it occupies in stationary or marine practice, with all of the appurtenances designed to facilitate the economical production of steam, and the means of housing or protecting the same. Upon this broad basis should be judged the efficiency of mechanical draft as a most important factor in modern steam-boiler practice. Its many advantages, as clearly proven by extended use and careful trial, will be presented at length in succeeding chapters.

It is proper here, however, to consider the influence, from a commercial standpoint, which the application of mechanical draft exerts upon the ultimate efficiency of a steam-boiler plant as measured by its aggregate first cost and the consequent fixed charges thereon. For this purpose there has been selected a plant of reasonable size of which the detailed cost is known. This plant, as illustrated in plan in Fig. 1, consists of 8 modern water-tube boilers, each of 200 horse-power nominal rating, set in pairs, making a total of 1,600 horse-power. A chimney is provided 8 feet in internal diameter by 180 feet high, of sufficient capacity to overcome the resistance of the economizers and produce the draft necessary for any probable forcing of the boilers. Two feed-water economizers are provided, through or around which the gases may be caused to pass on their way to the chimney. The boilers and economizers are enclosed in an independent boiler house, outside which stands the chimney. The detailed cost of that portion of the plant which concerns the present discussion is in round numbers as follows: —

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[1] Closed Stokeholds. Richard Sennett. Transactions of Institution of Naval Architects, 1886.

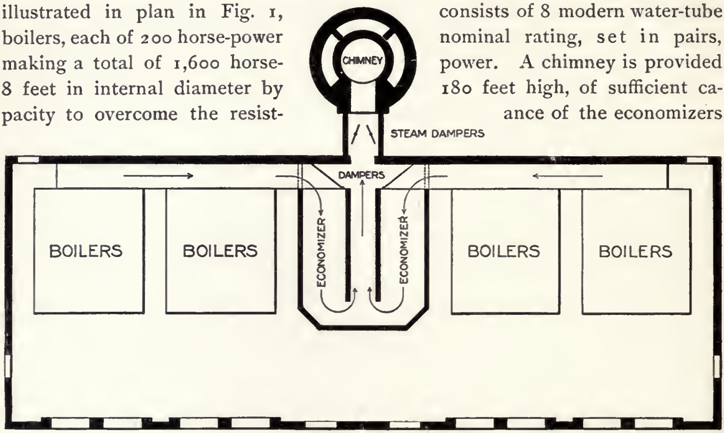

FIG. 1.

8 water-tube boilers of 200 horse-power each,	$25,000.00
2 feed-water economizers	7,000.00
Boiler and economizer setting and by-pass .	6,000.00
Automatic damper regulator and dampers .	300.00
Chimney, complete	9,000.00
Building, complete	11,000.00
	$58,300.00

Being taken in round numbers, these costs may be considered to represent a fair average for the vicinity of New England. The cost of the chimney includes the foundations necessary where the ground is stable, but there is always a possibility of considerable increased cost when the nature of the ground demands deeper or more extended foundations.

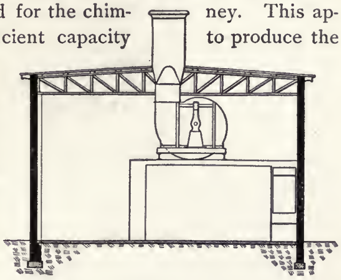

FIG. 2.

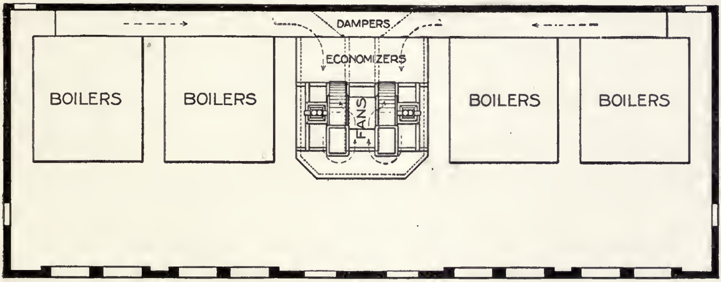

FIG. 3.

In Figs. 2 and 3 is shown the simplified arrangement which is possible when a mechanical draft apparatus is substituted for the chimney. This apparatus, of the induced type, is of sufficient capacity to produce the same maximum draft as the chimney, and may be readily placed on top of the economizers without occupying space otherwise valuable. The apparatus as shown in the illustrations is of the duplex type, consisting of two steel-plate fans placed side by side, each driven by a double-cylindered upright engine. Each fan as designed is capable of independently producing the draft for the entire plant, and may, therefore, be operated alone at near its maximum speed, or both fans may be driven at less speed to accomplish the same results. This duplex arrangement with fans in duplicate is not positively necessary except in cases where the work required is practically continuous, and where any delay from possible accident to the apparatus would cause great inconvenience. By means of an arrangement of dampers in the inlet connection between the fans, the hot gases drawn from the space beneath may be caused to pass through either of the two fans, while access may be had to the interior of the other without inconvenience from the heat. By means of a special form of draft regulator the fans are kept continuously in operation at just the speed necessary to produce the draft that, with different conditions of the fire, will maintain the steam pressure constant.

This is one of the most important features of such an arrangement, for, as the conditions of the fire change, it is obvious that the intensity of the draft and the amount of air supplied should change in like proportion. In a subsequent chapter are shown coincident record cards from draft and steam-pressure recording gauges, which display the conditions which are maintained by mechanical draft. The gases leaving the fans pass to a short connecting steel stack extending through the roof.

The same general arrangement of economizer, by-pass and dampers may be preserved; while the chimney and the space occupied thereby are rendered unnecessary. Such an apparatus, with its duplicate engines and fans, its automatic regulation and the short stack, can be installed complete, under ordinary conditions, for about $3,500. Evidently, the cost would be much less were the duplicate feature omitted, and only a single fan of the requisite capacity employed. As the special draft regulator takes the place of the damper regulator usually employed in connection with the chimney, the total economy in first cost effected by the introduction of the mechanical draft plant may be indicated thus; the saving of space occupied by the chimney being neglected: —

Chimney Draft.		*Mechanical Draft.*	
Cost of chimney . .	$9,000.00	Cost of fans, engines, draft regulator, and short stack, installed complete .	$3,500.00
Cost of damper regulator and dampers . .	300.00	Saving by use of mechanical draft	5,800.00
	$9,300.00		$9,300.00

It is thus evident that the mechanical draft apparatus here costs only about 38 per cent of the chimney and damper regulators. Had only a single fan of the requisite capacity been used instead of the relay arrangement shown, the cost of the mechanical draft apparatus would have been reduced to about $2,000, making its relative cost only about 22 per cent of the chimney and draft regulator, and the saving would amount to $7,300.

A still further reduction might have been secured by designing the plant so as to operate the boilers at somewhat above their rated capacity, as could be readily done by means of a mechanical draft apparatus, and in fact by means of the same apparatus upon which the cost has already been given. Although such reduction would most naturally be made in the rated capacity of each boiler, yet, for illustration, it would be simpler to consider the effect of omitting one of the boilers of the plant as designed. This would bring the rated capacity down to 1,400 horse-power, and would call upon the fans to only increase the steaming capacity of the other boilers by about 14 per cent above the normal. This would show an additional saving in first cost which may be thus presented: —

1,600 Nominal Horse-Power Plant.

Cost of 8 boilers . .	$25,000.00
Cost of settings, etc. .	6,000.00
Cost of building . .	11,000.00
	$42,000.00

1,400 Nominal Horse-Power Plant.

Cost of 7 boilers, . .	$21,875.00
Cost of settings, etc., about	5,500.00
Cost of building, about .	10,500.00
Saving by use of mechanical draft	4,125.00
	$42,000.00

This shows a possible supplementary saving on the entire plant of $4,125, which, in addition to the direct saving already shown to be effected by the introduction of mechanical draft, making no allowance for reduced expense on account of less space occupied, makes a total reduction of $9,925 to be credited to the account of the mechanical method. This saving is equal to more than the entire cost of chimney and damper regulator, and to nearly three times the first cost of the mechanical draft plant. Of course, the fixed charges for interest, taxes and insurance will be correspondingly reduced. Had this comparison been based upon the cost of a forced draft plant,— that is, one in which the air was forced rather than drawn through the fire,— the saving in the cost would have been shown to be even greater; for a single fan would probably have served the purpose, and, being employed to handle the air at atmospheric pressure, would not require to be as large as the induced draft plant, through which must pass the air and gases when nearly doubled in volume by their increase in temperature.

In any properly arranged power plant there should be no question whatever as to the opportunity to utilize the exhaust steam from the fan engine. This may, and in a plant of proper design would, be used for heating buildings, for feed-water heating, or would be eventually saved by condensation and return to the boilers. Under such conditions the actual cost of the steam used in producing draft would be reduced to practically nothing.

In this estimate the cost of the land occupied by the chimney or the invested capital sufficient to earn the rental rate thereon has been neglected; but if this were an electric lighting or power plant situated in the heart of a large city, this would also form an important factor in the economy of the mechanical draft system. The space occupied by the chimney, measuring the latter by the area of the enclosing rectangle, amounts to 495 square feet, which, at a nominal rate of, say, $2 per square foot, would cost $990. A similar allowance made for the decreased space resulting from a reduction in the number of boilers while maintaining the same steaming capacity, would show a saving at $2 per foot upon 480 square feet, or a total of $960. Of course these items depend entirely upon the actual cost of the land, and will vary greatly according to its location.

The total net saving in first cost of a single plant, under the given conditions, may be thus summarized: —

By omission of chimney and damper regulator .	$5,800.00
By reduction in number of boilers	4,125.00
By saving in space occupied by chimney . .	990.00
By saving in space occupied by boiler omitted .	960.00
	$11,875.00

This saving is made possible by the expenditure of $3,500 for the mechanical draft apparatus; that is, the saving is nearly three and a half times the expenditure necessary to secure it. Had it been desired to secure the desired 1,600 horse-power by increasing the steaming capacity of the boilers above the mere 14 per cent here estimated upon, the saving would have been greater.

The reduction of $11,875 in the cost would indicate an annual saving in fixed charges of about $831 to $890, according as the aggregate of interest, taxes and insurance is taken at 7 or 7½ per cent. This amount would, under conditions of the best economy, be practically sufficient to cover the cost of operating the mechanical draft apparatus, provided no attempt was made to utilize the exhaust steam. When to this is added the increased convenience of mechanical draft, its positive character, its ready adaptability, its independence of climatic conditions, and its instant response to any demand for increased steam supply, the account stands decidedly to its credit. Therefore any further saving, as for instance in the cost of the fuel burned, is clear gain over and above any expenditure which may have been made on account of the introduction of this method.

In another light, however, the saving in fuel cost may be considered as regards the amount per ton which the price would have to be reduced in order to cover the cost of operating the fan in case all of the exhaust steam is thrown away. We may take for illustration the same 1,600 horse-power plant. Upon the commercial horse-power rating of 34.5 pounds of water evaporated from and at 212°, this would indicate a total hourly evaporation of 55,200 pounds of water. If 11 pounds of water were evaporated per pound of coal under these conditions, this would call for $\frac{55,200}{11} = 5,018$ pounds of coal per hour, or 2.509 tons of 2,000 pounds each. Estimating on 312 working days of 10 hours each, this requires 7,828 tons per year. A saving of only 15 cents per ton effected by the installation of a mechanical draft plant would indicate a reduction in the fuel account of $1,174.20 per year, an amount sufficient under ordinary conditions to pay the full cost of operating the fan, provided it was impossible to utilize the exhaust steam. Whatever net amount can be saved per ton, over and above this amount, can be credited to the mechanical draft and added to the reduction in fixed charges. When it is known that a saving in cost of coal of from $50 to $75 per week may be readily shown in a plant of this size and character, the economical possibilities in the way of mechanical draft are evident beyond question. Such a weekly saving is equivalent to an annual reduction in the fuel cost of from $2,600 to $3,900, certainly a most excellent return on an investment of only $3,500, or, rather, on an original net saving of $11,875. In the preceding chapter has been instanced the case of a boiler plant of 1,005 horse-power nominal rating, in which the fuel saving due to the introduction of mechanical draft amounted to $126 per week. This is equivalent to an annual saving of $6,552, or far more than the original cost of the draft plant.

Although by no means all of the factors concerned in the increased ultimate efficiency of steam boilers have been here considered, nevertheless those presented have been sufficient to clearly point to the economical advantages of mechanical draft as relating to its first cost. Additional influences upon the economy effected by this method of draft production will be presented in succeeding chapters.

CHAPTER VII.

RATE OF COMBUSTION.

Rate of Combustion. — The rate at which fuel is consumed in steam-boiler practice is usually expressed in pounds per hour per square foot of grate surface. Evidently, this rate must vary greatly according to the conditions governing in a given case. Although the draft is by far the most important factor, yet the amount of coal burned per unit of area also depends, to a certain extent, upon the total area and arrangement of the grate surface, the method of firing, the kind of fuel, and other factors of less importance.

In the early days of the steam boiler the rate of combustion, like the speed of the first steam engines, was extremely low. The grates were large, the draft comparatively light, and forcing of the boilers less common than at the present day. The general rates of combustion, as given by Rankine[1] for various types of boilers and conditions of draft, are presented in Table No. 83.

Table No. 83. — Rates of Combustion.

Conditions.	Method of Producing Draft	Pounds of Coal per Square Foot of Grate per Hour.
Slowest rate of combustion in Cornish boilers	Chimney.	4
Ordinary rates in Cornish boilers	Chimney.	10
Ordinary rates in factory boilers	Chimney.	12 to 16
Ordinary rates in marine boilers	Chimney.	16 to 24
Quickest rates of complete combustion of dry coal, the supply of air coming through the grate only,	Chimney.	20 to 23
Quickest rates of complete combustion of caking coal, with air holes above the fuel to the extent of 1-36 area of grate,	Chimney.	24 to 27
Locomotives	Blast pipe or fan.	40 to 120

Since these figures were first given the rates have increased, so that 15 to 20 pounds is not at all uncommon in factory boilers, under exceptionally strong draft; while 25 to 40 pounds is the usual marine practice, and 60 to 125 pounds are burned in the boilers of torpedo boats. In fact, modern steam-engineering practice is constantly looking toward higher rates of combustion as an accompaniment to higher steam pressures and engine speeds, in the attempt to attain increased efficiency.

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[1] The Steam Engine and Other Prime Movers. W. J. M. Rankine. London, 1885.

Mr. W. S. Hutton[1] states that "the application of forced draft to a furnace affords a means of obtaining a higher rate of combustion of fuel per square foot of grate surface per hour than is conveniently obtainable with natural draft. The rate of combustion obtained in practice varies, with the intensity of the draft, from 30 to 200 pounds of coal per square foot of fire-grate surface per hour. A moderate rate of forced combustion is from 35 to 50 pounds of coal per square foot of grate per hour."

Relation of Grate Surface to Heating Surface. — The ratio existing between the areas of grate and heating surface in various types of boilers has already been presented in Table No. 72. The influence of this ratio on the evaporation has also been indicated. As the economy of the boiler is usually expressed in the number of pounds of water evaporated per pound of coal, it is evident that, the surface ratio and the rate of combustion also being known, the relative capacity of the boiler can be determined and expressed in pounds of water evaporated per square foot of heating surface. In fact, this should be the ultimate basis of comparison rather than the rate of combustion; for the influence of the latter is dependent upon the surface ratio, which may vary considerably even in boilers of the same type. The results of this influence are shown in Table No. 84, in which, with a constant fuel efficiency of 10 pounds of water

Table No. 84. — Rates of Evaporation per Square Foot of Heating Surface.

SURFACE RATIO.	POUNDS OF COAL PER HOUR PER SQUARE FOOT OF GRATE.								
	5	10	15	20	25	30	35	40	45
1 : 25	2.00	4.00	6.00	8.00	10.00	12.00	14.00	16.00	18.00
1 : 30	1.67	3.33	5.00	6.66	8.33	10.00	11.67	13.33	15.00
1 : 35	1.43	2.86	4.29	5.72	7.15	8.58	10.00	11.44	12.87
1 : 40	1.25	2.50	3.75	5.00	6.25	7.50	8.75	10.00	11.25
1 : 45	1.11	2.22	3.33	4.44	5.55	6.66	7.77	8.88	10.00
1 : 50	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
1 : 55	0.91	1.82	2.73	3.64	4.55	5.46	6.36	7.27	8.17
1 : 60	0.83	1.67	2.50	3.33	4.17	5.00	5.63	6.67	7.50

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[1] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

evaporated from and at 212° per pound of coal, the evaporation per square foot of heating surface has been calculated for different rates of combustion under different surface ratios. Of course this table is purely theoretical, and applies only when the efficiency is 10 pounds, but it indicates relative values.

It is evident that, with a given boiler, it is a comparatively simple matter to so change the setting as to permit of the use of grates of a different area, thus altering the surface ratio; and, further, that a reduction of grate surface with the maintenance of the same rate of combustion is relatively equivalent to an increase in the heating surface without change of the original grate. A reduction of grate area, without corresponding increase in the rate of combustion, such that the total amount of fuel consumed remains the same, must of necessity reduce the rate of evaporation per square foot of heating surface, and consequently the capacity of the boiler. Under such conditions the evaporative efficiency may be improved, but the boiler be actually incapable of performing its previous amount of work. It is by such a combination of circumstances that the unprincipled advocates of certain so-called coal-saving devices have sometimes been able to show an increased evaporation per pound of fuel, which, while satisfactory in itself, was not secured under the ordinary working conditions, and might possibly have been as easily secured without the device. The capacity may be maintained, even with reduced grate area, by correspondingly increasing the rate of combustion. The limit to such arrangements is naturally set by the amount of fuel which can be fired and maintained in good condition per square foot of grate; while the evaporative capacity of the heating surface is limited by the ability of the steam bubbles to readily escape from its surface, which ability in turn is largely dependent upon the arrangement of the surface.

The possibilities of increased capacity are evidenced in the statement of Mr. W. S. Hutton,[1] that "with an efficient system of forced draft 1 indicated horse-power may generally be economically developed from 2 square feet of heating surface of marine return tubular boilers fired with good coal. The proportion of heating surface to fire-grate surface may be 45 to 1. Small tubes are more effective with forced draft than large tubes, and the smoke tubes may be at least one-quarter less in diameter than the tubes used for natural draft."

Economy of High Rates of Combustion. — It has been shown in the preceding chapter that with a *constant* surface ratio the evaporative efficiency of the boiler decreases as the rate of combustion is increased. But an increase of the rate, while the surface ratio remains the same, is in effect an increase in the total quantity of coal consumed. The more the coal consumed the more the air required; hence the logical deduction, substantiated by experience, that the efficiency will be reduced. For the larger volume of gases travelling at higher velocity will impart relatively less heat to the exposed surfaces and enter the chimney at a higher temperature. But in practice certain factors affect these theoretical considerations, so that, as evidenced by statements which follow, the rate of combustion upon a given grate may be greatly increased without adverse economic effect.

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[1] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

It is to be noted that when the surface ratio is constant the rate of combustion becomes practically a direct measure of the total amount of fuel consumed. If, however, the ratio be changed, as for instance by reducing the grate area, the total consumption can only be maintained by increasing the rate of combustion per square foot. Under this condition it is certainly evident that at least no greater amount of air will be required per pound of coal. In fact, experience shows that ordinarily the amount of air required will actually be reduced. This naturally results in an increase in the efficiency. Upon this fact rests one of the important advantages of mechanical draft, for by its means may be produced the intensity of draft requisite to high combustion rates.

The conditions under which this increased efficiency can be secured must, however, be clearly understood. An increase in the rate of combustion, when it is accompanied by a greater coal consumption, as would be the case where the surface ratio remains constant, is not always conducive to economy. But, when the total amount of coal consumed remains the same, and the increased combustion rate is secured by a reduction of grate area and corresponding increase in the surface ratio, a higher efficiency is the natural result. This is shown by Table No. 73, the figures in which give relative values.

As there indicated, the efficiency of the fuel, or the water evaporated from and at 212° per pound of coal, is 10.23 pounds for a stationary boiler, where the rate of combustion is 30 pounds per square foot of grate and the surface ratio is 30. This is equivalent to burning $30 \div 30 = 1$ pound of coal per square foot of heating surface per hour. If the surface ratio were 50, and the coal consumption per square foot of heating surface remained the same,— namely, 1 pound,— the rate per square foot of grate would be $50 \times 1 = 50$ pounds. The table shows that with these conditions of surface ratio and rate of combustion the evaporation would be 10.67 pounds, an increase of about 4 per cent. The high rates are chosen for illustration only because they avoid interpolation in the table; but the same principle holds throughout. For instance, a rate of 25 pounds per square foot of grate, with a ratio of 50, gives about 9 per cent higher efficiency than a rate of 15 pounds and a ratio of 30, although the coal consumption per square foot of heating surface is the same.

The principal if not the sole cause for this increase in efficiency is to be found in the decreased supply of air which is required per pound of coal when the rate of combustion per square foot of grate is raised. The reason of the decreased requirement appears evident in the fact that the higher rate of combustion necessitates a deeper fire, and that the air supplied is, therefore, compelled to come in contact with a greater amount of fuel, and afforded a better opportunity to promote perfect combustion. The intensity of the fire is increased, its temperature is higher, more heat is radiated to the exposed boiler surfaces, and more is taken up by the gases. Furthermore, the diminished superficial area of the grate and of the exposed interstices between the fuel necessitates a higher velocity to secure the admission of a given volume of air. This increased velocity in turn requires greater draft or air pressure. The volume at a given temperature passing through the coal is proportional to the velocity, but the pressure varies as the square of the velocity. Therefore, if a given grate be reduced one-half, and the rate of combustion doubled, so as to maintain the same total consumption, the same volume of air would have to travel through the exposed interstices at twice the velocity. But the pressure or vacuum would be four times as great, and, as a consequence, the air would be forced or drawn into spaces between the fuel which it could not reach under lesser impelling force. Much more intimate contact and distribution are the results. Less free oxygen passes through the fuel bed unconsumed, and for a given supply of air a higher efficiency of the fuel is attained. But the high pressures necessary to such results seldom exist or are attainable where a chimney is depended upon for the production of the draft. In fact, the present rates of combustion common in factory practice are such, largely because of the inability of a chimney of moderate height and cost to economically provide draft of sufficient intensity for higher rates. For this reason forced or mechanical draft has always been considered in a sense separate and apart from chimney draft, and until recently has been regarded principally in the capacity of producing draft pressures beyond the limits of the ordinary chimney. Consequently, the effects of increased draft and higher combustion rates have been attributed directly to the use of mechanical means; and, in fact, experience has shown that these effects can be economically attained only by the employment of such means. This fact it is the object of this work to establish by an impartial statement of such experience.

Thus Rankine wrote nearly forty years ago that "in furnaces where the draft is produced by means of a blast pipe, like those of locomotive engines or by means of a fan, the quantity of air required for dilution, although it has not yet been exactly ascertained, is certainly much less than that which is required in furnaces with chimney drafts; and there is reason to believe that on an average it may be estimated at about *one-half* of the air required for combustion."

After stating that a "high [smoke] pipe increases economy of combustion, due to more energetic combination of oxygen and fuel, Engineer-in-Chief Melville[1] states that 'this has been repeatedly shown with moderate forced draft.'"

Hutton[2] bases his estimate of increased efficiency with mechanical draft upon the decreased amount of air required therewith, and the resulting higher furnace temperature. Accepting 24 pounds of air as necessary per pound of coal under natural draft, he shows the furnace temperature to be 2,926°, with perfect combustion, giving an efficiency of 66 per cent. Accepting 18 pounds as required under forced draft, the furnace temperature is shown to be 3,686°, and the efficiency 76 per cent.

He states that "more complete combustion, giving a higher temperature, may be obtained in a furnace with forced than with natural draft. The heating surfaces of the boiler are also more efficient because there is a greater difference in the temperature of the water surface and fire surface of the plates forming the heating surfaces. As the rate of transfer of heat varies as the difference in temperature of the water on one side of the plate and that of the fuel gases on the other side; the greater the difference, the greater the amount of heat which will pass through a unit of heating surface in a given time."

This subject has not been so carefully investigated as its importance warrants, but it is a well-established fact that, under ordinary conditions, a decreased amount of air is required with rapid rates of combustion, such as obtain in mechanical draft practice. Owing to the influence of surface ratios, kind of fuel, construction of the furnace, and the like, any absolute basis of comparison is, however, practically impossible.

But the importance of mechanical draft as an adjunct to economical combustion is appreciated by all progressive engineers. Thus Prof. Carpenter[3] concisely and conservatively states that "one of the requisites of economy that is required is high temperature in the furnace. This, on the other hand, means a small supply of air, but an amount which is sufficient to support combustion. To secure perfect combustion with a small amount of air requires an intimate distribution of the air and fuel. This of necessity requires a strong draft, which possibly can be more cheaply produced by mechanical means than by heating the chimney." He then proceeds to show by the results of an actual test "that for that plant, at least, a gain of a good many horse-power would have been possible by the substitution of mechanical for natural draft; provided the heat discharged from the flue could have been prevented and utilized."

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[1] Machinery of the New Vessels of the United States Navy. George W. Melville. Transactions Society of Naval Architects and Marine Engineers. 1893.

[2] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

[3] Wastes from Boiler Management. R. C. Carpenter. Machinery, June, 1895.

Recent extended experience with American boilers and coals, as shown by the following quotations, points clearly to the substantial maintenance under proper operation of the evaporative efficiency with widely varying rates of combustion. Mr. F. R. Low, in the recent report of the Committee on Data of the National Electric Light Association,[1] states: "I have plotted the water evaporated per hour per square foot of heating surface, and the pounds of water evaporated from and at 212° per pound of combustible, as determined by 30 different tests on Babcock & Wilcox boilers, and proved that practically as good results are obtained at over 5 pounds per square foot of heating surface as at 1.75 pounds, and the intermediate tests show no dependence on the rate of evaporation. I have plotted in the same way all of the tests of which I could find a record, and in this wide range still no evidence is apparent of any dependence of the boilers represented on the rate of evaporation within the range covered. This means that the variation of efficiency of boilers within the range here comprised is less than the variations due to different firing, etc.

"Wm. H. Bryan, in a paper read before the Engineers' Club of St. Louis, gives the data and tests in which a battery of horizontal tubular boilers were forced to more than double their rating with only the following improvement of their efficiency: [In brief the tabulated results show the coal per square foot of grate surface to range between 18.074 and 43.68, that per square foot of heating surface between 0.332 and 0.803, the water evaporated per square foot of heating surface between 2.43 and 5.235, and the evaporation per pound of combustible from and at 212°, between 9.27 and 8.827. In percentage of rated capacity the range is between 100.2 and 219.83, while the efficiency percentage of heat utilized varies between 76.38 in the former and 68.83 in the latter.] Here is a battery of boilers which were forced to nearly double their rated capacity with a decrease of only 6 per cent in their efficiency, and which could doubtless have been diminished to one-third of their rated capacity with a less impairment still. In other words, with good management and an adaptation to conditions, these boilers would have taken care of a maximum load six times the minimum without suffering extremely." These statements are not to be overlooked when considering the application of mechanical draft.

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[1] The Engineering Record. New York, June 26, 1897.

Clark[1] states that "the proportion of surplus air required appears to diminish as the rate of combustion and the general temperature in the furnace are increased," and presents the results here given in Table No. 85 as evidence of the truth of this statement. Although the surplus air with the Longridge boiler appears extremely low, yet the experiments of Whitham, given in Table No. 86, although conducted under special conditions, tend to confirm its probability.

Table No. 85.—Surplus Air with Different Rates of Combustion.

KIND OF BOILER.	Coal consumed per Square Foot of Grate per Hour. Pounds.	Surplus Air. Per cent.
Cornish	2 to 4	100
Delabèche and Playfair .	10 to 16	25 to 50
Longridge	20 and upwards.	9¾

Although it is commonly accepted that the air supply with chimney draft is about 300 cubic feet (approximately 24 pounds per pound of coal where the combustion rate is from 10 to 15), yet there is in practice a wide variation from this standard. Thus the tests of Messrs. Donkin and Kennedy, already presented in Table No. 16, show a range between 17.3 pounds and 39.2 pounds per pound of coal; that is, an excess of 56 to 260 per cent. In marine practice, with mechanical draft, a combustion rate of 30 to 40 pounds may be easily maintained with a supply of 225 cubic feet of air per pound of coal. Under the influence of mechanical draft, whereby the volume may be readily controlled, the air supply requisite for successful combustion is much reduced below that with chimney draft. How much this reduction may be must depend upon the conditions,—character of fuel, rate of combustion, etc.

Table No. 86.—Air Supply with Different Rates of Combustion on a Wilkinson Automatic Mechanical Stoker.

Buckwheat Coal burned per Hour per Square Foot of Stoker Grate. Pounds.	Air Theoretically Required to burn One Pound of. Buckwheat Coal. Cubic Feet.	Air Actually Supplied to burn One Pound of Buckwheat Coal. Cubic Feet.	Percentage of Excess or Deficiency of Air Supplied.
12.0	125	232	+ 85.6
18.0	125	157	+ 25.6
25.2	125	132	+ 5.6
32.5	125	123	− 1.6
41.5	125	111	− 11.2
45.4	125	111	− 11.2

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[1] The Steam Engine. D. K. Clark. London, &c., 1890.

Greater care in the distribution of the air and in maintaining the condition of the fire is necessary to a successful reduction in the air supply. This is very clearly evidenced by the remarkable results of tests by Mr. J. M. Whitham[1] upon automatic mechanical stokers. Those relating to the Wilkinson stoker, under different rates of combustion, are presented in Table No. 86. It is to be noted that there is an excess of air up to a combustion of 30 pounds of dry coal per hour per square foot of grate. An evaporative efficiency of 11.69 pounds of water from and at 212° per pound of combustible was recorded when burning 45.4 pounds, while the table shows that there was an actual deficiency in the air supply. The natural explanation of maintained efficiency with this air supply is to be found in the construction of the stoker. It consists of a number of hollow grate bars, to the ends of which the air is admitted and through numerous small holes in the upper surfaces of which it escapes. The air is thus thoroughly diffused throughout the fire; there is practically perfect contact of air and fuel, and consequent combustion with the minimum of waste gases.

The deductions of Mr. D. K. Clark, already given in the preceding chapter, regarding the relation of grate area, heating surface, water and fuel, serve to confirm the statement that an increased rate of combustion does not entail decreased efficiency when the total consumption remains constant. Regardless of any reduction in the air supply required per pound of coal when the rate is increased, he showed by the results of extended experiment[2] upon locomotive boilers that the efficiency remained practically constant when the rate of combustion increased and the total evaporation proceeded in the ratio of the square of the surface ratio. The results, with values calculated therefrom, are presented in Table No. 87. It is to be noted that under the experimental conditions the capacity of the group of boilers, as measured in pounds of water

Table No. 87.—Effects of Different Surface Ratios and Rates of Combustion.

Designation Groups of Tests.	Coke Consumed per Square Foot of Grate per Hour. Pounds.	Average Ratio of Heating Surface to Grate Surface.	Coke Consumed per Square Foot of Heating Surface per Hour. Pounds.	Water Evaporated per Square Foot of Heating Surface per Hour. Pounds.	Water Evaporated per Pound of Coke. Pounds.
A	42.7	52	0.82	7.4	9.0
B	55.0	66	0.83	7.6	9.1
C	86.0	72	1.19	10.6	8.9
D	126.0	90	1.40	12.5	8.92

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[1] Experiments with Automatic Mechanical Stokers. J. M. Whitham. Transactions American Society of Mechanical Engineers, Vol. XVII.

[2] The Steam Engine. D. K. Clark. London, &c., 1890.

evaporated per hour per square foot of heating surface, increased from 7.4 to 12.5, or 69 per cent, while the efficiency remained substantially constant. It is a perfectly reasonable inference that had the capacity been maintained constant, the efficiency would have increased with the rate of combustion.

Further and more direct evidence that with a given total coal consumption the efficiency of the boiler rises as the rate of combustion is increased, is presented by the tests of M. Burnat[1] upon a French boiler with grates of three different areas. The principal items in these results are given in Table No. 88, which shows that with practically the same quantity of coal consumed per hour on the three grates the efficiency increased as the grate area was diminished in the ratios of 7.26 pounds, 7.54 pounds and 7.79 pounds of water per pound of coal, although in the third case a larger supply of air was provided.

Table No. 88. — Results of Performance of French Boiler with Grates of Different Areas.

Area of Grate.	Air at 62° Fahr. per Pound of Coal.	Average Temperature.		Coal Consumed.		Residue.	Water per Pound of Coal from and at 212°.
		Feed Water.	Gas at Dampers.	Per Hour.	Per Hour per Square Foot of Grate.		
Square Feet.	Cubic Feet.	Degrees.	Degrees.	Pounds.	Pounds.	Per cent.	Pounds.
24.70	161	124°	576°	125	5.28	16.5	7.26
12.37	164	124	612	127	11.00	18.7	7.54
9.03	180	117	570	124	14.74	19.0	7.79

Similar confirmatory results[2] with grates 6 feet and 4 feet long are presented in Table No. 89, showing that higher efficiency and rapidity of evaporation were obtained from the shorter grate with about equal quantities of coal per hour.

Table No. 89. — Effect of Length of Grate upon Efficiency and Rapidity of Evaporation.

ITEMS.		LENGTH OF GRATE.	
		4 Feet.	6 Feet.
Total coal per hour	pounds,	400	414
Coal per hour per square foot of grate	pounds,	14	23
Water at 212° evaporated per pound of coal . . .	pounds,	10.10	10.91

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[1] Bulletin de la Société Industrielle de Mulhouse, Vol. XXX. 1859–60.

[2] The Steam Engine. D. K. Clark. London, &c., 1890.

The inference must of necessity be drawn from the preceding that an increase in the surface ratio and a corresponding increase in the rate of combustion will under proper conditions result in raising the efficiency. Under the present conditions of boiler design and the arrangements existing in most boiler plants, the simplest means of securing the desired results would appear to lie in the introduction of feed-water, or air heaters, or similar devices of such proportions that the gases resulting from the more rapid combustion produced by mechanical means will be prevented from passing to the fan at too high a temperature.

Thickness of Fire. — It is commonly accepted that to economically burn an increased quantity of coal per square foot per hour it is necessary to increase the thickness of the layer of fuel. Comparative tests with different thicknesses of fire in a marine boiler, by Messrs. Richardson and Fletcher,[1] showed, per Table No. 90, that the efficiency increased with the thickness of the fire.

Table No. 90. — Efficiency of Thick Fires.

ITEMS.		THICKNESS OF FIRE.		
		9-inch.	12-inch.	14-inch.
Coal consumed per square foot of grate per hour,	pounds,	27	27	27
Water evaporated per pound of coal, as supplied at 212°,	pounds,	10.77	11.23	11.54

There are conditions, however, where the thinner fire may prove the more economical, as where the rate of combustion is such that a heavy fire, fed at long intervals and given but little attention is compared with a thinner fire fed more frequently and run under better management. Evidently, the stronger draft is required with the thicker fire; but this, as has already been pointed out, should be an element in the increased efficiency because of the greater pressure which causes more intimate contact of air and fuel, as with mechanical draft.

As stated by Mr. W. S. Hutton,[2] "A thick fire is necessary for economical combustion with forced draft. It should not, in a general way, be less than 10 inches thick, and it should not be allowed to burn down to a less thickness than 7 inches before stoking. A thin fire causes loss from the entrance through the fuel of an excessive supply of air. The stronger the draft the thicker must the fire be."

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[1] Report on the Boiler and Smoke Prevention Trials, conducted at Wigan, 1869.

[2] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

CHAPTER VIII.

DRAFT.

Definition. — More or less confusion exists in the use of the term "draft" in boiler practice because of its double meaning. As usually employed, it refers to the difference in pressure between the external air and the gases as they leave the boiler; although, as related to the combustion of the fuel, it should properly apply to the difference between the under- and over-grate pressures. In either application it indicates the intensity or force of the draft, and is generally measured in inches of water by means of a draft gauge. The term "draft" is, however, sometimes employed as a measure of the volume or weight of the gases passing through the fire. As the readings of a draft gauge give no direct indication of their volume, the quantity of air or gases must be determined by other means.

In the case of a chimney, the maximum intensity of draft exists only with the maximum temperature of the gases; but after the temperature reaches about 600° Fahr. their density decreases more rapidly than their velocity increases, so that the weight of air supplied is a maximum at about this temperature. As the draft is almost universally measured by the difference in pressure, expressed either in inches of water or in weight per unit of area, the term will be here employed as indicating the intensity or force with which it acts. This difference in pressure, whether it be the result of creating a plenum in the ashpit or a partial vacuum in the boiler furnace, is always necessary to produce the flow of air through the fuel whereby combustion is maintained. It is evident, therefore, that the draft or pressure difference and the velocity or air flow are interdependent.

Relation of Pressure and Velocity. — As the laws which govern the movement of gases are the same as those which apply to liquids in motion, their application can be most readily illustrated by means of a liquid, which has visible substance. If a vessel with vertical sides, as indicated in Fig. 4, be filled with water at 50° Fahr. to the level A, the total pressure upon the bottoms will be equal to the weight of the entire quantity of water. If the area of the base be 100 square inches, and the total weight of water be 1,500 pounds, the pressure per square inch will be $\frac{1,500}{100} = 15$ pounds. This indicates that each column of water having 1 square inch for its base, and the distance AB $= h$ for its height, weighs 1.5 pounds. As the weight of water per cubic foot at 50° is 62.409 pounds, and consequently 0.0361 per cubic inch, it is also evident that the distance AB must be $\frac{15}{0.0361} = 415.3$ inches $= 34.6$ feet. This depth of water, h, producing the given pressure per square inch, is known as the total head, and in this case is also the hydrostatic, or pressure head. Obviously the pressure exerted is directly proportional to the head or depth of water. For the cross-section of the vessel remaining constant, any change in the depth of the water must result in a coincident change in the total weight of water which presses upon the bottom.

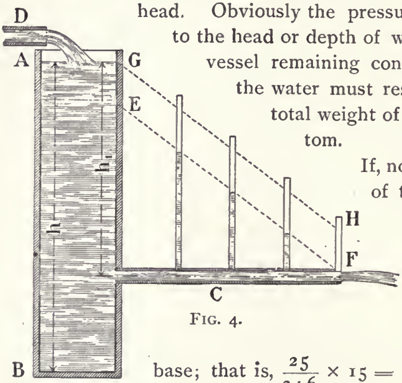

Fig. 4.

If, now, a pipe C be inserted in the side of the vessel at such a height above the base that the distance h_1 is 25 feet, it is evident that the pressure per square inch of cross-section of the vessel at this point will be $\frac{25}{34.6}$ of that upon the base; that is, $\frac{25}{34.6} \times 15 = 10.85$ pounds. In other words, it will be equivalent to the head of water multiplied by its density, as clearly shown by the equations, —

$$p = hd \quad \text{and} \quad h = \frac{p}{d}$$

In which $p =$ pressure.
$h =$ head.
$d =$ density.

This pressure is transmitted to the water in the end of the pipe at its junction with the vessel. If, now, the vessel be arranged to receive a continual supply of water through D, such that its level A will be kept constant, notwithstanding the outward flow through the pipe C, the head h_1 will continue the same, as will also the pressure which is exerted upon the water in the end of the pipe. By the action of gravity the water seeks to escape through the pipe, and its effect or pressure is exactly proportional to the head or depth h_1.

If there were no friction whatever in the pipe the total head would be rendered effective for producing flow, and the speed or velocity of that flow would be exactly that which would finally result if a body under the action of gravity had freely fallen the distance measured by the head h. Therefore, its velocity is determinable by the well-known formula for falling bodies, viz.: —

$$v = \sqrt{2gh}$$

In which $v =$ velocity in feet per second.
$h =$ the head in feet.
$g =$ acceleration due to gravity $= 32.16$.

As it has already been shown that $h = \frac{p}{d}$, this formula, as applied to the velocity of movement of fluids, may take the form —

$$v = \sqrt{2g\frac{p}{d}}$$

In practice the walls of the pipe under consideration would restrict the freedom of flow, and a portion of the total head or pressure would have to be expended in overcoming the resistance. This resistance is naturally greatest at the part of the pipe farthest removed from the outlet, for the freedom of flow increases as the water nears the end of the pipe where there is no resistance; the atmospheric pressure at this point being balanced by that upon the upper surface of the water in the vessel. If the pipe should be provided with a series of small open-topped gauges, as shown, they would respectively indicate, by the heights of water within them, the resistances which exist at the different points. The decreasing heights of these columns as they approach the end of the pipe is evidence of the decreasing resistance. The regularity of this fall is indicated by the dotted line EF, while the fall in the total head available at each of these points is represented by the line GH. Since these lines are parallel, the vertical distance between them, which represents the portion of the head utilized for producing velocity, is evidently constant.

As the total head represents the sole means of producing movement of the water, whatever portion of this head is used for overcoming resistance reduces by just so much the amount that remains available for the production of velocity. That portion of the total head which is thus employed to overcome resistance is known as the *pressure head*, while that remaining and utilized for the production of velocity is designated as the *velocity head*.

Evidently, if the pipe be of uniform diameter and the water be considered non-compressible, the velocity must be uniform, and hence the velocity or pressure expended for producing that velocity must also be uniform. This confirms the evidence of the parallel dotted lines in the figure.

With a constant total head any increase in the length of the pipe naturally increases the pressure head and consequently reduces the velocity head. If, however, the pipe be entirely removed, leaving only the orifice, the pressure head will be practically eliminated and the total head will become the velocity head. The actual velocity through the opening will be very slightly less than that calculated by the formula, owing to the slight friction of the water in passing through. But the volume passing through an opening in a flat plate will be considerably less than that which would be calculated by multiplying the area of the opening by the velocity, or rate of flow. This is due to the fact that the stream contracts as it leaves the opening, so that its minimum area, which is at a short distance from the orifice, becomes only about two-thirds of the area of the opening. The effect of different forms of opening will be discussed in their relation to the efflux of air.

Efflux of Air. — As the pressure is dependent upon both the height and the density of the fluid, it is evident that for a given pressure the less the density the greater the height of the column. But the law of falling bodies recognizes the fact that it is the distance fallen through and not the weight of the body that determines its velocity. Therefore, the less dense a body the higher the column required to produce a given pressure and the greater the velocity of discharge. From this it is evident that the velocity of a gas issuing under a given pressure would be greater than that of a liquid under the same conditions. And conversely, the more dense the fluid issuing at a given velocity the greater must have been the pressure to produce that velocity.

In the case of a liquid, the atmospheric pressure upon the inlet and outlet of a containing vessel is balanced and the actual height or head may be actually measured. But air is invisible, and there is no tangible distinction in substance between that producing the pressure and that constituting the surrounding atmosphere.

The pressure of the atmosphere is due to the weight of the air, and, for any area, is to be measured by the weight of a column of air having the given area as a base and a height equal to that of the atmosphere. But this height cannot be accurately determined, and, furthermore, the density of the air decreases in geometric ratio as the distance from the earth increases. For the purposes of calculation, however, the practical equivalent of such a column may be determined by assuming the air to be of uniform density throughout and the column of such a height as to weigh the same and to produce the same effective pressure per unit of area.

Under the standard conditions of barometric pressure of 29.921 inches, the atmospheric pressure is 14.69 pounds per square inch, or 2,115.36 pounds per square foot. At this pressure a cubic foot of dry air at 50° has a density of 0.077884 pounds. Consequently a homogeneous column $\frac{2,115.36}{0.077884} = 27,160$ feet high, having a base of one square foot area, would weigh 2,115.36 pounds and exert this pressure upon the given area.

If air under this head were to be allowed to flow freely into a vacuum, the velocity, per the formula, would be —

$$\begin{aligned} v &= \sqrt{2g \times 27,160} \\ &= \sqrt{64.32 \times 27,160} \\ &= 1,321.7 \text{ feet per second.} \end{aligned}$$

In the case of air flowing into a vacuum the total head is actual, although here reduced for simplicity to that of a homogeneous column. But under any other conditions both the pressure head and the velocity head are purely ideal. The fact that a given air pressure exists in a reservoir does not indicate the existence of an actual column of air of known density. But the height of such an ideal column is readily determinable by calculation from the simple equation, —

$$h = \frac{p}{d}$$

and it is this ideal height that is used in calculation.

If, as is frequently the case, the pressure is expressed in inches of water, as indicated by the balanced height of a column of that liquid in a water gauge, this may be readily transformed into the height of the equivalent column of air. Thus, if the pressure difference in inches of water be represented by H, and the equivalent head of air in feet by h, the value of h will be, —

$$h = \frac{\text{density of water} \times H}{12 \times \text{density of air}}$$

then at a temperature of 50° Fahr. —

$$\begin{aligned} h &= \frac{62.409 \times H}{12 \times 0.077884} \\ &= 66.77H \end{aligned}$$

If this value be employed in the formula for velocity, it becomes —

$$\begin{aligned} v &= \sqrt{64.32 \times 66.77H} \\ &= 65.5\sqrt{H} \end{aligned}$$

from which may be approximately determined the velocity of efflux of air under any given pressure difference expressed in inches of water. As the value of H

is dependent upon the temperature of both air and water, more particularly the former, it is evident that the value of the constant applies only under the stated conditions; and the value of v must be considered as only approximate where the formula is employed under other conditions without suitable corrections therefor.

But for more refined calculation additional factors must be taken into consideration. For simplicity the atmospheric pressure and humidity may be considered constant, and differences of pressure rather than absolute pressures are employed. In the case of air, which is compressible by pressure and expansible by heat, the density must vary with the pressure; and a change in the temperature may have a most important influence. The effect of increased density, which may be produced by the pressure, is to decrease the ideal velocity head. For if $h = \frac{p}{d}$, it is evident that an increase in d must for a given pressure reduce the value of h. A similar influence is exerted by the temperature, for by an increase in temperature the density is decreased, and hence the value of h is increased. The velocity being dependent solely upon the ideal head, it is of the utmost importance that comparisons be reduced to the same conditions of pressure, temperature and density.

If the pressure is to be expressed in ounces per square inch, which is readily reducible to inches of water, the formula —

$$v = \sqrt{2g\frac{p}{d}}$$

when applied under the conditions of —

g = acceleration due to gravity = 32.16,
p = pressure in ounces per square inch,
d = density or weight of one cubic foot of dry air at 50° temperature, and under atmospheric pressure = 0.077884 pounds,

becomes —

$$v = \sqrt{64.32 \times \frac{p \times 144}{16 \times 0.077884 \times \frac{235 + p}{235}}}$$

In this form it is evident that the density varies with the pressure, for p being expressed in ounces, and the atmospheric pressure of 14.69 pounds being equivalent to 235 ounces, the density which exists at any given pressure p is $0.077884 \times \frac{235 + p}{235}$

The formula reduces to —

$$v = \sqrt{64.32 \times \frac{p \times 9}{0.077884} \times \frac{235}{235 + p}}$$

$$= \sqrt{\frac{1,746,659 \times p}{235 + p}}$$

The formula, when the pressure is expressed in inches of water, and allowance is made for the increased density of the air due to the pressure, takes the form —

$$v = \sqrt{\frac{1,746,659 \times h}{406.7 + h}}$$

Both formulæ thus take into account the compression of the air due to its pressure, but make no allowance for change of temperature during discharge.

From these two basis formulæ the velocities given in Tables Nos. 91 and 92 have been calculated; the pressures being expressed in ounces per square inch in the former and in inches of water in the latter. In both tables the air is assumed to be dry and of a temperature of 50° Fahr. As it has already been shown that the velocity varies as the square root of the head, and that the head $h = \frac{p}{d}$, it must be evident that any decrease in the density of the air due to a higher temperature must for the same pressure increase the value of h, and consequently increase the velocity. Furthermore, if the velocity and consequently the head are to be maintained constant under varying temperatures, the pressure must decrease proportionately to the density as affected by the temperature.

In considerations of density the absolute temperatures only are concerned. Thus, the absolute temperature at 50° Fahr. is $461 + 50 = 511°$, while that at, say, 550° Fahr. is $461 + 550 = 1,011$. Hence the relative density at 550° Fahr. is $\frac{511}{1,011} = 0.5054$, and consequently the relative volume of the same weight of air, expanded by the increase in temperature, is $\frac{1,011}{511} = 1.978$.

The relative values thus calculated have been incorporated in the second and third columns of Table No. 93. From the second column it is evident that if the velocity with which air at 50°, under a pressure of 2½ ounces per square inch, issues from an orifice be 8,135.7 feet per minute, as per Table No. 91, then that at which it issues under the same pressure, but at a temperature of 150°, will be $1.09 \times 8,135.7 = 8,867.9$ feet. Column 3 shows, on the other hand, that if a velocity of 8,867.9 feet per minute be observed under the above conditions, it must have been due to a pressure of $0.84 \times 2.5 = 2.1$ ounces per square inch.

Table No. 91. — Velocity Created, Volume Discharged and Horse-Power Required when Air under a Given Pressure in Ounces per Square Inch is Allowed to Escape into the Atmosphere.

Pressure in Ounces.	Velocity of Dry Air at 50° Temperature Fahr. Escaping into the Atmosphere through any Shaped Orifice in any Pipe or Reservoir in which the Given Pressure is Maintained.		Volume of Air in Cubic Feet which may be Discharged in One Minute through an Orifice having an Effective Area of Discharge of One Square Inch.	Horse-Power required to move the Given Volume of Air under the Given Conditions of Discharge.
Per Square Inch.	In Feet per Second.	In Feet per Minute.		
⅛	30.47	1,828.4	12.69	0.00043
¼	43.08	2,585.0	17.95	0.00122
⅜	52.75	3,165.1	21.98	0.00225
½	60.90	3,653.8	25.37	0.00346
⅝	68.07	4,084.0	28.36	0.00483
¾	74.54	4,472.6	31.06	0.00635
⅞	80.50	4,829.7	33.54	0.00800
1	86.03	5,161.7	35.85	0.00978
1⅛	91.22	5,473.4	38.01	0.01166
1¼	96.13	5,768.0	40.06	0.01366
1⅜	100.80	6,047.9	42.00	0.01575
1½	105.25	6,315.2	43.86	0.01794
1⅝	109.52	6,571.3	45.63	0.02022
1¾	113.64	6,817.6	47.34	0.02260
1⅞	117.58	7,055.0	49.00	0.02505
2	121.41	7,284.4	50.59	0.02759
2⅛	125.11	7,506.7	52.13	0.03021
2¼	128.70	7,722.2	53.63	0.03291
2⅜	132.20	7,931.8	55.08	0.03568
2½	135.59	8,135.7	56.50	0.03852
2⅝	138.91	8,334.4	57.88	0.04144
2¾	142.14	8,528.3	59.22	0.04442
2⅞	145.29	8,717.6	60.54	0.04747
3	148.38	8,902.8	61.83	0.05058
3⅛	151.40	9,084.0	63.08	0.05376
3¼	154.36	9,261.5	64.32	0.05701
3⅜	157.26	9,435.4	65.52	0.06031
3½	160.10	9,606.1	66.71	0.06368
3⅝	162.89	9,773.3	67.87	0.06710
3¾	165.63	9,938.0	69.01	0.07058
3⅞	168.33	10,099.6	70.14	0.07412

Table No. 91.—Velocity, Volume and Horse-Power, &c.—Concluded.

Pressure.	Velocity per Second.	Velocity per Minute.	Volume.	Horse-Power.
4	170.98	10,258.6	71.24	0.07771
4¼	176.15	10,568.8	73.39	0.08507
4½	181.16	10,869.5	75.48	0.09264
4¾	186.03	11,161.5	77.51	0.1004
5	190.76	11,445.5	79.48	0.1084
5¼	195.37	11,722.0	81.40	0.1166
5½	199.86	11,991.5	83.24	0.1249
5¾	204.25	12,254.8	85.10	0.1335
6	208.53	12,511.9	86.89	0.1422
6½	216.82	13,009.3	90.34	0.1602
7	224.77	13,486.4	93.66	0.1788
7½	232.42	13,945.4	96.84	0.1981
8	239.80	14,387.9	99.92	0.2180
8½	246.92	14,815.4	102.88	0.2385
9	253.83	15,229.6	105.76	0.2596
9½	260.52	15,631.0	108.55	0.2812
10	267.00	16,020.4	111.25	0.3034
10½	273.32	16,399.3	113.88	0.3261
11	279.70	16,768.1	116.45	0.3493
11½	285.46	17,127.6	118.94	0.3730
12	291.30	17,478.2	121.38	0.3972
12½	297.01	17,820.6	123.75	0.4219
13	302.59	18,155.2	126.06	0.4470
13½	308.04	18,482.4	128.35	0.4726
14	313.38	18,802.7	130.57	0.4986
14½	318.61	19,116.3	132.75	0.5250
15	323.73	19,423.6	134.89	0.5518
15½	328.75	19,725.0	136.98	0.5791
16	333.68	20,020.7	139.03	0.6067
16½	338.51	20,310.8	141.05	0.6347
17	343.26	20,595.8	143.03	0.6631
17½	347.93	20,875.8	144.97	0.6919
18	352.52	21,151.0	146.88	0.7211
18½	357.03	21,421.6	148.76	0.7506
19	361.46	21,687.8	150.61	0.7804
19½	365.83	21,949.7	152.43	0.8107
20	370.13	22,207.5	154.22	0.8412

Table No. 92.—Velocity Created when Air under a Given Pressure in Inches of Water is Allowed to Escape into the Atmosphere.

Pressure in Inches of Water, per Square Inch.	Velocity of Dry Air at 50° Temperature Escaping into the Atmosphere through any Shaped Orifice in any Pipe or Reservoir in which the Given Pressure is Maintained.		Pressure in Inches of Water, per Square Inch.	Velocity of Dry Air at 50° Temperature Escaping into the Atmosphere through any Shaped Orifice in any Pipe or Reservoir in which the Given Pressure is Maintained.	
	In Feet per Second.	In Feet per Minute.		In Feet per Second.	In Feet per Minute.
0.1	20.72	1,243.3	2.6	105.33	6,320.0
0.2	29.30	1,758.0	2.7	107.33	6,439.7
0.3	35.84	2,150.4	2.8	109.28	6,557.0
0.4	41.43	2,485.6	2.9	111.21	6,672.3
0.5	46.31	2,778.7	3.0	113.09	6,785.5
0.6	50.73	3,043.5	3.1	114.95	6,896.8
0.7	54.78	3,287.0	3.2	116.77	7,006.3
0.8	58.56	3,513.5	3.3	118.57	7,114.1
0.9	62.10	3,726.1	3.4	120.34	7,220.2
1.0	65.45	3,927.2	3.5	122.08	7,324.7
1.1	68.64	4,118.4	3.6	123.80	7,427.7
1.2	71.68	4,301.0	3.7	125.49	7,529.3
1.3	74.60	4,476.1	3.8	127.16	7,629.4
1.4	77.41	4,644.5	3.9	128.80	7,728.2
1.5	80.12	4,806.9	4.0	130.43	7,825.7
1.6	82.73	4,963.9	4.25	134.40	8,064.1
1.7	85.27	5,116.1	4.5	138.26	8,295.4
1.8	87.73	5,263.7	4.75	142.00	8,520.1
1.9	90.12	5,407.3	5	145.65	8,738.8
2.0	92.45	5,547.1	5.25	149.20	8,951.8
2.1	94.72	5,683.4	5.5	152.66	9,159.7
2.2	96.94	5,816.5	5.75	156.05	9,362.8
2.3	99.11	5,946.4	6	159.35	9,561.2
2.4	101.23	6,073.6	6.25	162.59	9,755.4
2.5	103.30	6,198.1	6.50	165.76	9,945.8

In the preceding discussion, pressures have been considered to be above the atmosphere; but in all cases pressure differences only have been expressed. The results apply as accurately to pressures below the atmosphere, and practically may be so applied; for there is no appreciable difference, for instance, between the velocity of air issuing into the atmosphere from a reservoir under 4 ounces pressure per square inch, and that passing from the atmosphere into a reservoir in which a vacuum of 4 ounces below the atmosphere is maintained.

Influence of Form of Orifice. — The form of the orifice through which the air passes under pressure has practically no effect upon the velocity. But the volume of air discharged is largely dependent upon the character of the opening. As already stated, the stream of any fluid escaping through an orifice in a thin plate is reduced in size just beyond the opening. This is due to the fact that the direction of the molecules of the fluid is changed in passing through the orifice. Experiments have shown that this *vena contracta*, or contracted vein, at a distance from the orifice equal to half its diameter, experiences its maximum contraction, and that in the case of water its diameter is about 0.8 of that of the orifice. The orifice being round, the area of the stream varies as the square of the diameter; hence the area becomes $0.8^2 = 0.64$ of that of the orifice. This quantity is known as the *coefficient of contraction*. The coefficient of efflux is the product of the coefficient of contraction and the coefficient of velocity. The latter quantity, in the case of water flowing through a round orifice, is 0.97 without appreciable error, and may be considered as unity in the case of orifices having a higher coefficient of contraction than that for a round orifice. This latter coefficient, and consequently the volume discharged, may be greatly increased by substituting for the round orifice pipes or openings of such form as to render easier the outflow of the fluid; hence the importance of careful attention to this matter.

The coefficient of efflux of air through an orifice in a thin plate varies somewhat with the diameter of the orifice and the ratio of the pressures. Thus with an orifice 1 centimeter in diameter Weisbach found the coefficient to vary from 0.555, when the greater pressure was 1.05 times that of the lesser, to 0.788 when the ratio was at 2.15 to 1. With an opening 2.14 centimeters in diameter, the coefficient, with the ratio of pressures of 1.05 to 1, was 0.558, and 0.723 when the ratio was 2.01 to 1.

The continuation of the orifice in the form of a pipe serves to increase the coefficient of efflux, as is further evident from the experiments of Weisbach. With a tube 1 centimeter in diameter and 2 centimeters long, the coefficient was found to be 0.730 for a pressure ratio of 1.05 to 1, and 0.830 when the ratio was 1.30 to 1. A short pipe, 1 centimeter in diameter and 1.6 centimeters long, with its inlet well rounded off, thereby rendering easier the entrance of the air, showed a coefficient which averaged 0.976 under different pressure ratios between 1.24 and 2.14. A complete nozzle consisting of a conical tube with an angle of convergence of 6°, which was 145 centimeters long, 1 centimeter in diameter at the outlet and 3.8 centimeters in diameter at the inlet, which was rounded off, gave for a ratio of 1.08 to 1 a coefficient of 0.932, and for a ratio of 2.16 to 1 a coefficient of 0.984.

In round numbers the coefficient of efflux, when the pressure differences are comparatively small, as in the case of a fan, may be taken as follows: —

For an orifice in a *thin plate*	0.56
For a *short cylindrical pipe*	0.75
For a *rounded-off conical mouthpiece*	0.98
For a *conical pipe* whose angle of convergence is about 6°,	0.92

Of course the proper coefficient must be applied in any given case. But it is simple to calculate the volume of air which at a given velocity would pass in a stream of known effective area. The results of such calculations are given in Table No. 91, in which is indicated the volume of air passing per minute at the stated velocity in a stream of one square inch effective area. How much larger than this area the orifice should be in order to make the contracted vein one square inch in area must of course depend upon the form of the opening.

In most cases the weight of air moved by any given means is the final desideratum; therefore, due allowance must be made for variations in temperature. The necessity of this is shown in Table No. 91, in columns 4 and 5, which indicate respectively the relative weights of air moved at a given velocity and the relative velocities necessary to move the same weight under various temperatures.

Work Required to Move Air. — The theoretical amount of energy, as expressed in foot-pounds, which is expended in moving a given volume of air, is measured by the product of the distance moved and the total resistance which is overcome. Thus, as per Table No. 91, under a pressure of 8 ounces per square inch the velocity of issuing air is 14,387.9 feet per minute. If the effective area of discharge be 6 square inches, the total pressure becomes $6 \times 8 = 48$ ounces, or 3 pounds; and the work done is $14{,}387.9 \times 3 = 43{,}163.7$ foot-pounds. As this work is accomplished in one minute, it equals $\frac{43{,}163.7}{33{,}000} = 1.31$ H. P.

In this manner the theoretical horse-power has been calculated for one square inch of effective area under the temperature, pressure and velocity conditions of Table No. 91, and therein incorporated in the last column. How much more than this theoretical amount will be actually required must depend upon the efficiency of the machine or device by which the result is accomplished. Evidently, with a constant velocity due to a constant head, $h = \frac{p}{d}$, the actual pressure must vary directly as the density of the air and inversely as its absolute temperature. This has already been explained and is presented in column 3 of Table No. 93. Therefore, if the velocity remains constant, the power required to overcome the resistance must be exactly proportional to the relative pressure.

Table No. 93.—Influence of the Temperature of Air upon the Conditions of its Movement.

Temperature in Degrees. Fahr. 1	Relative Velocity Due to the Same Pressure. 2	Relative Pressure Necessary to Produce the Same Velocity. 3	Relative Weight of Air Moved at the Same Velocity. 4	Relative Velocity Necessary to Move the Same Weight of Air. 5	Relative Pressure Necessary to Produce the Velocity to Move the Same Weight of Air. 6	Relative Power Necessary to Move the Same Volume of Air at the Same Velocity. 7	Relative Power Necessary to Move the Same Weight of Air at the Velocity in Column 5 and the Pressure in Column 6. 8
30°	0.98	1.04	1.04	0.96	0.96	1.04	0.92
40	0.99	1.02	1.02	0.98	0.98	1.02	0.96
50	1.00	1.00	1.00	1.00	1.00	1.00	1.00
60	1.01	0.98	0.98	1.02	1.02	0.98	1.04
70	1.02	0.96	0.96	1.04	1.04	0.96	1.08
80	1.03	0.94	0.94	1.06	1.06	0.94	1.12
90	1.04	0.93	0.93	1.08	1.08	0.93	1.17
100	1.05	0.91	0.91	1.10	1.10	0.91	1.21
125	1.07	0.87	0.87	1.15	1.15	0.87	1.32
150	1.09	0.84	0.84	1.20	1.20	0.84	1.43
175	1.11	0.81	0.81	1.24	1.24	0.81	1.55
200	1.14	0.78	0.78	1.29	1.29	0.78	1.67
225	1.16	0.75	0.75	1.34	1.34	0.75	1.80
250	1.18	0.72	0.72	1.39	1.39	0.72	1.93
275	1.20	0.69	0.69	1.44	1.44	0.69	2.07
300	1.22	0.67	0.67	1.49	1.49	0.67	2.22
325	1.24	0.65	0.65	1.54	1.54	0.65	2.36
350	1.26	0.63	0.63	1.59	1.59	0.63	2.51
375	1.28	0.61	0.61	1.63	1.63	0.61	2.66
400	1.30	0.59	0.59	1.68	1.68	0.59	2.82
425	1.32	0.58	0.58	1.73	1.73	0.58	2.99
450	1.34	0.56	0.56	1.78	1.78	0.56	3.17
475	1.35	0.55	0.55	1.83	1.83	0.55	3.35
500	1.37	0.53	0.53	1.88	1.88	0.53	3.53
525	1.39	0.52	0.52	1.93	1.93	0.52	3.72
550	1.41	0.51	0.51	1.98	1.98	0.51	3.92
575	1.43	0.49	0.49	2.03	2.03	0.49	4.12
600	1.44	0.48	0.48	2.08	2.08	0.48	4.33
625	1.46	0.47	0.47	2.13	2.13	0.47	4.54
650	1.48	0.46	0.46	2.18	2.18	0.46	4.75
675	1.49	0.45	0.45	2.22	2.22	0.45	4.93
700	1.51	0.44	0.44	2.27	2.27	0.44	5.15
725	1.52	0.43	0.43	2.32	2.32	0.43	5.38
750	1.54	0.42	0.42	2.37	2.37	0.42	5.62
775	1.56	0.41	0.41	2.42	2.42	0.41	5.86
800	1.57	0.40	0.40	2.47	2.47	0.40	6.10

As a consequence, the values given in column 7 are identical with those in column 3. The velocity being constant, the volume discharged is also constant, but its relative weight is as shown in column 4.

If it be desired to pass through the same orifice a constant weight of air, its velocity must necessarily vary directly with its increase in absolute temperature, for its density correspondingly decreases. The velocity necessary to move the same weight is produced under each different temperature by the relative pressure shown in column 6. The pressure thus necessary to produce this velocity must at constant temperature evidently increase with the square of the velocity, and at other temperatures must coincidently decrease inversely with the absolute temperature; that is, proportionately to the density.

For illustration take the case of air at a temperature of 300°. Per the table, column 5, the velocity necessary to move the same weight as at 50° is relatively 1.49. For its production this would call for a relative pressure of $1.49^2 = 2.22$ at 50°, but at the temperature of 300° the pressure required to produce the given velocity is, per column 3, only 0.67 of that required at 50°. Hence the relative pressure required at 300° to produce the velocity necessary to move the same weight of air is relatively $2.22 \times 0.67 = 1.49$ times that which is necessary to produce the movement of the same weight, but less volume, at 50°. Under these conditions of moving the same weight at different temperatures, the relative power required is evidently the product of the factors in column 5 and in column 6, for it is represented by the product of the pressure into the velocity. Upon this basis column 8 has been calculated. From this is evident the fact that the work performed is not proportional to the weight of the air moved, but to the distance through which the resistance is overcome.

The power required to move air by means of a fan is of great importance in the consideration of any system of mechanical draft. Other things equal, and friction neglected, the power required to drive a fan increases as the cube of its speed. For the pressure increases as its square, the velocity obviously increases as its speed, and the work done is the product of these two factors. Furthermore, the speed remaining constant, the volume also remains constant, while the weight of air moved and the power required both decrease in proportion to the density of the air; that is, inversely as its absolute temperature. The subject of fans will be discussed at length in the chapter on Mechanical Draft. The cause for the enormous waste of energy in the movement of air by a chimney is, as explained in the chapter on Chimney Draft, due to the fact that the energy is not directly applied, as with a fan, but that the air movement is secured by the expenditure of heat in raising the temperature, and reducing the density of the gases so that gravity may act to produce the flow.

Movement of Air in Pipes. — Air in its movement in pipes or conduits is resisted by its friction upon their interior surfaces. This resistance of friction is proportional to the surface with which it comes in contact; that is, directly to the length and inversely to the diameter of the pipe. It also varies as the square of the velocity, and is expressed by Weisbach's[1] well-known formula —

$$h = f\frac{l}{d} \times \frac{v^2}{2g}$$

In which f = coefficient of resistance of friction, to be determined by experiment.
l = length of pipe.
d = diameter of pipe.
v = velocity of the air.
g = acceleration due to gravity = 32.16.

The value of f evidently controls the result, and must depend both upon the material and character of construction of the pipe. Assuming the pipe to be of galvanized iron, carefully made and erected with all internal laps extending in the direction of the air movement, the following formulæ, with constants in round numbers, have been deduced from that previously given: —

$$p = \frac{lv^2}{25{,}000d} \qquad v = \sqrt{\frac{25{,}000dp}{l}}$$

$$l = \frac{25{,}000dp}{v^2} \qquad d = \frac{lv^2}{25{,}000p}$$

In all of which p = loss of pressure in ounces per square inch.
v = velocity in feet per second.
l = length of pipe in feet.
d = diameter of pipe in inches.

Taking the weight of one cubic foot of air, in round numbers, as 0.08 pounds, and expressing the area of the pipe by A, the horse-power lost in friction in a pipe 100 feet long may be determined by the formula —

$$\text{H. P.} = \frac{pAv}{8{,}800}$$

By means of the formulæ for loss of pressure and horse-power lost in friction, Table No. 94 has been calculated for pipes of various diameters, all 100 feet long, with air travelling at different velocities expressed in feet per minute. No

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[1] Mechanics of Engineering. Julius Weisbach, Ph. D. Translated by Eckley B. Coxe, A. M. New York, 1878.

Table No. 94.—Pressure and Horse-Power Required to Compensate for the Friction of Air Passing through Pipes.

Velocity of Air in Feet per Minute	DIAMETER OF PIPE.																			
	1-inch.		2-inch.		3-inch.		4-inch.		5-inch.		6-inch.		7-inch.		8-inch.		9-inch.		10-inch.	
	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.
100	0.011	0.0000	0.006	0.0000	0.004	0.0000	0.003	0.0000	0.002	0.0000	0.002	0.0000	0.002	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000
200	0.044	0.0000	0.022	0.0000	0.015	0.0000	0.011	0.0001	0.009	0.0001	0.007	0.0001	0.006	0.0001	0.006	0.0001	0.005	0.0001	0.004	0.0001
300	0.100	0.0000	0.050	0.0001	0.033	0.0001	0.025	0.0002	0.020	0.0002	0.017	0.0003	0.014	0.0003	0.012	0.0004	0.011	0.0004	0.010	0.0004
400	0.178	0.0001	0.088	0.0002	0.059	0.0003	0.044	0.0004	0.036	0.0005	0.030	0.0006	0.025	0.0007	0.022	0.0008	0.020	0.0009	0.018	0.0011
500	0.278	0.0002	0.139	0.0004	0.092	0.0006	0.069	0.0008	0.056	0.0010	0.046	0.0012	0.040	0.0014	0.035	0.0017	0.031	0.0019	0.028	0.0021
600	0.400	0.0004	0.200	0.0007	0.133	0.0011	0.100	0.0014	0.080	0.0018	0.067	0.0021	0.057	0.0025	0.050	0.0029	0.044	0.0032	0.040	0.0036
700	0.544	0.0006	0.272	0.0011	0.181	0.0017	0.138	0.0023	0.109	0.0028	0.091	0.0034	0.078	0.0040	0.068	0.0045	0.060	0.0051	0.054	0.0057
800	0.711	0.0008	0.356	0.0017	0.237	0.0025	0.178	0.0034	0.142	0.0042	0.119	0.0051	0.102	0.0059	0.089	0.0067	0.079	0.0076	0.071	0.0085
900	0.900	0.0012	0.450	0.0024	0.300	0.0036	0.225	0.0048	0.180	0.0060	0.150	0.0072	0.129	0.0084	0.112	0.0096	0.100	0.0108	0.090	0.0120
1,000	1.111	0.0017	0.556	0.0033	0.370	0.0049	0.278	0.0066	0.222	0.0083	0.185	0.0099	0.159	0.0116	0.139	0.0132	0.123	0.0148	0.111	0.0165
1,100	1.344	0.0022	0.672	0.0044	0.448	0.0066	0.338	0.0088	0.269	0.0110	0.224	0.0132	0.192	0.0154	0.168	0.0176	0.149	0.0198	0.134	0.0220
1,200	1.600	0.0029	0.800	0.0057	0.533	0.0086	0.400	0.0114	0.320	0.0143	0.267	0.0171	0.229	0.0200	0.200	0.0230	0.178	0.0256	0.160	0.0286
1,300	1.878	0.0036	0.939	0.0073	0.626	0.0109	0.469	0.0145	0.376	0.0182	0.313	0.0218	0.268	0.0254	0.235	0.0290	0.209	0.0327	0.188	0.0363
1,400	2.178	0.0045	1.089	0.0090	0.726	0.0136	0.544	0.0181	0.436	0.0227	0.363	0.0272	0.311	0.0317	0.282	0.0363	0.242	0.0408	0.218	0.0453
1,500	2.500	0.0056	1.250	0.0112	0.833	0.0167	0.625	0.0223	0.500	0.0279	0.417	0.0335	0.357	0.0390	0.312	0.0446	0.278	0.0502	0.250	0.0558
1,600	2.844	0.0068	1.422	0.0135	0.948	0.0203	0.711	0.0271	0.569	0.0338	0.474	0.0406	0.406	0.0473	0.356	0.0541	0.316	0.0609	0.284	0.0677
1,700	3.211	0.0081	1.605	0.0162	1.070	0.0244	0.803	0.0325	0.642	0.0406	0.535	0.0487	0.459	0.0568	0.401	0.0649	0.357	0.0731	0.321	0.0812
1,800	3.600	0.0096	1.800	0.0192	1.200	0.0289	0.900	0.0385	0.720	0.0482	0.600	0.0578	0.514	0.0674	0.450	0.0771	0.400	0.0867	0.360	0.0964
1,900	4.011	0.0113	2.006	0.0227	1.337	0.0340	1.003	0.0453	0.802	0.0567	0.669	0.0680	0.573	0.0793	0.501	0.0907	0.446	0.1020	0.401	0.1133
2,000	4.444	0.0132	2.222	0.0264	1.481	0.0397	1.111	0.0529	0.889	0.0661	0.741	0.0793	0.635	0.0925	0.556	0.1058	0.493	0.1190	0.444	0.1322
2,200	5.378	0.0176	2.689	0.0352	1.793	0.0528	1.344	0.0704	1.076	0.0880	0.896	0.1161	0.683	0.1232	0.672	0.1408	0.597	0.1583	0.538	0.1760
2,400	6.400	0.0228	3.200	0.0457	2.133	0.0685	1.600	0.0914	1.280	0.1142	1.067	0.1371	0.914	0.1600	0.800	0.1728	0.711	0.2056	0.640	0.2284
2,600	7.511	0.0290	3.756	0.0581	2.504	0.0871	1.877	0.1162	1.502	0.1452	1.252	0.1743	1.073	0.2033	0.939	0.2324	0.835	0.2614	0.751	0.2904
2,800	8.711	0.0363	4.356	0.0726	2.904	0.1088	2.178	0.1451	1.742	0.1814	1.452	0.2177	1.244	0.2539	1.089	0.2902	0.968	0.3265	0.871	0.3628
3,000	10.000	0.0446	5.000	0.0892	3.333	0.1339	2.500	0.1785	2.000	0.2231	1.667	0.2677	1.429	0.3123	1.250	0.3569	1.111	0.4016	1.000	0.4462
3,200	11.378	0.0541	5.689	0.1083	3.792	0.1625	2.844	0.2166	2.276	0.2708	1.896	0.3249	1.625	0.3790	1.422	0.4332	1.263	0.4873	1.138	0.5415
3,400	12.844	0.0649	6.422	0.1299	4.281	0.1949	3.211	0.2598	2.569	0.3247	2.141	0.3897	1.835	0.4546	1.606	0.5237	1.427	0.5845	1.284	0.6495
3,600	14.400	0.0770	7.200	0.1542	4.800	0.2313	3.600	0.3084	2.880	0.3855	2.400	0.4626	2.057	0.5397	1.800	0.6168	1.600	0.7039	1.440	0.7710
3,800	16.044	0.0906	8.022	0.1814	5.349	0.2720	4.011	0.3627	3.209	0.4534	2.674	0.5441	2.292	0.6347	2.006	0.7254	1.783	0.8171	1.604	0.9068
4,000	17.778	0.1058	8.889	0.2115	5.926	0.3173	4.444	0.4230	3.556	0.5288	2.963	0.6346	2.540	0.7403	2.222	0.8461	1.975	0.9518	1.778	1.0576
4,200			9.800	0.2448	6.553	0.3673	4.900	0.4897	3.920	0.6121	3.267	0.7346	2.800	0.8570	2.450	0.9794	2.178	1.1019	1.960	1.2243
4,400			10.705	0.2815	7.175	0.4223	5.353	0.5631	4.282	0.7038	3.569	0.8446	3.059	0.9854	2.676	1.1261	2.379	1.2669	2.141	1.4077
4,800			12.800	0.3655	8.533	0.5484	6.400	0.7310	5.120	0.9138	4.267	1.0964	3.657	1.2793	3.200	1.4620	2.844	1.6448	2.560	1.8275
5,200			15.022	0.4747	10.015	0.6971	7.511	0.9294	6.008	1.1618	5.007	1.3941	4.292	1.6265	3.756	1.8588	3.339	2.0912	3.004	2.3236
5,600			17.422	0.5804	11.615	0.8707	8.711	1.1608	6.969	1.4510	5.807	1.7412	4.978	2.0314	4.356	2.3216	3.872	2.6118	3.484	2.9021
6,000			20.000	0.7139	13.333	1.0710	10.000	1.4278	8.000	1.7847	6.667	2.1416	5.714	2.4986	5.000	2.8555	4.444	3.2125	4.000	3.5694

Table No. 94.—Pressure and Horse-Power Required to Compensate for the Friction of Air Passing through Pipes.—Continued.

Velocity of Air in Feet per Minute.	DIAMETER OF PIPE.																			
	11-inch.		12-inch.		13-inch.		14-inch.		16-inch.		18-inch.		20-inch.		22-inch.		24-inch.		26-inch.	
	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.
100	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.001	0.0000	0.000	0.0000	0.000	0.0000
200	0.004	0.0001	0.004	0.0002	0.003	0.0002	0.003	0.0002	0.003	0.0002	0.002	0.0002	0.002	0.0003	0.002	0.0003	0.002	0.0003	0.002	0.0003
300	0.009	0.0005	0.008	0.0005	0.008	0.0006	0.007	0.0006	0.006	0.0007	0.006	0.0008	0.005	0.0009	0.005	0.0010	0.004	0.0011	0.004	0.0012
400	0.016	0.0012	0.015	0.0013	0.014	0.0014	0.013	0.0015	0.011	0.0017	0.010	0.0019	0.009	0.0021	0.008	0.0023	0.007	0.0025	0.007	0.0027
500	0.025	0.0023	0.023	0.0025	0.021	0.0027	0.020	0.0029	0.017	0.0033	0.016	0.0037	0.014	0.0041	0.013	0.0045	0.012	0.0050	0.011	0.0054
600	0.036	0.0039	0.033	0.0043	0.031	0.0046	0.029	0.0050	0.025	0.0057	0.022	0.0064	0.020	0.0071	0.018	0.0079	0.017	0.0086	0.015	0.0093
700	0.049	0.0062	0.045	0.0068	0.041	0.0074	0.039	0.0079	0.034	0.0091	0.030	0.0102	0.027	0.0113	0.025	0.0125	0.023	0.0136	0.021	0.0147
800	0.064	0.0093	0.059	0.0102	0.055	0.0109	0.051	0.0118	0.044	0.0135	0.040	0.0152	0.036	0.0169	0.032	0.0186	0.029	0.0203	0.027	0.0220
900	0.081	0.0132	0.075	0.0145	0.069	0.0157	0.064	0.0169	0.056	0.0193	0.050	0.0217	0.045	0.0241	0.041	0.0265	0.037	0.0289	0.035	0.0313
1,000	0.101	0.0182	0.092	0.0198	0.085	0.0215	0.079	0.0231	0.069	0.0264	0.062	0.0297	0.056	0.0330	0.051	0.0364	0.046	0.0397	0.043	0.0430
1,100	0.122	0.0242	0.112	0.0264	0.103	0.0286	0.096	0.0308	0.084	0.0352	0.075	0.0396	0.067	0.0440	0.061	0.0484	0.056	0.0528	0.052	0.0572
1,200	0.145	0.0314	0.133	0.0343	0.123	0.0371	0.114	0.0400	0.100	0.0457	0.089	0.0512	0.080	0.0571	0.073	0.0628	0.067	0.0685	0.062	0.0742
1,300	0.170	0.0399	0.156	0.0437	0.144	0.0472	0.134	0.0508	0.117	0.0581	0.104	0.0654	0.094	0.0726	0.085	0.0799	0.078	0.0873	0.072	0.0944
1,400	0.198	0.0499	0.181	0.0544	0.167	0.0589	0.156	0.0635	0.136	0.0726	0.121	0.0816	0.109	0.0907	0.099	0.0998	0.091	0.1088	0.084	0.1179
1,500	0.227	0.0613	0.208	0.0669	0.192	0.0725	0.179	0.0781	0.156	0.0892	0.139	0.1004	0.125	0.1115	0.114	0.1227	0.104	0.1339	0.096	0.1450
1,600	0.259	0.0735	0.237	0.0812	0.219	0.0880	0.203	0.0948	0.178	0.1083	0.158	0.1218	0.142	0.1354	0.129	0.1469	0.119	0.1624	0.109	0.1760
1,700	0.292	0.0893	0.268	0.0974	0.247	0.1055	0.229	0.1137	0.207	0.1299	0.178	0.1461	0.161	0.1624	0.146	0.1786	0.134	0.1948	0.124	0.2110
1,800	0.327	0.1060	0.300	0.1156	0.278	0.1253	0.257	0.1351	0.225	0.1542	0.200	0.1735	0.180	0.1927	0.164	0.2120	0.150	0.2313	0.139	0.2506
1,900	0.365	0.1247	0.334	0.1360	0.308	0.1472	0.287	0.1587	0.251	0.1814	0.223	0.2040	0.201	0.2267	0.182	0.2494	0.167	0.2720	0.154	0.2945
2,000	0.404	0.1454	0.370	0.1586	0.341	0.1719	0.317	0.1851	0.278	0.2115	0.247	0.2380	0.222	0.2644	0.202	0.2908	0.185	0.3173	0.171	0.3437
2,200	0.489	0.1936	0.448	0.2111	0.413	0.2287	0.384	0.2463	0.336	0.2815	0.299	0.3167	0.269	0.3519	0.244	0.3871	0.224	0.4223	0.207	0.4575
2,400	0.582	0.2513	0.533	0.2741	0.492	0.2970	0.457	0.3198	0.400	0.3455	0.356	0.4112	0.320	0.4569	0.291	0.5026	0.267	0.5483	0.246	0.5939
2,600	0.683	0.3195	0.626	0.3485	0.578	0.3776	0.537	0.4066	0.468	0.4647	0.417	0.5228	0.376	0.5809	0.341	0.6390	0.313	0.6971	0.289	0.7552
2,800	0.792	0.3990	0.726	0.4353	0.670	0.4716	0.622	0.5079	0.544	0.5804	0.484	0.6530	0.436	0.7255	0.396	0.7981	0.363	0.8706	0.335	0.9432
3,000	0.909	0.4908	0.833	0.5354	0.769	0.5800	0.714	0.6245	0.625	0.7140	0.556	0.8031	0.500	0.8923	0.455	0.9816	0.417	1.0708	0.385	1.1601
3,200	1.034	0.5956	0.948	0.6498	0.875	0.7039	0.813	0.7581	0.711	0.8664	0.632	0.9747	0.569	1.0830	0.517	1.1913	0.474	1.2996	0.438	1.4079
3,400	1.168	0.7144	1.070	0.7794	0.988	0.8441	0.917	0.9093	0.827	1.0475	0.714	1.1690	0.642	1.2990	0.584	1.4289	0.535	1.5588	0.494	1.6882
3,600	1.309	0.8481	1.200	0.9252	1.114	1.0023	1.029	1.0800	0.900	1.2335	0.800	1.4078	0.720	1.5420	0.655	1.6962	0.600	1.8504	0.557	2.0046
3,800	1.459	0.9974	1.337	1.0871	1.234	1.1880	1.146	1.2695	1.003	1.4508	0.891	1.6342	0.802	1.8135	0.729	1.9949	0.669	2.1742	0.617	2.3760
4,000	1.616	1.1734	1.481	1.2691	1.367	1.3749	1.270	1.4807	1.111	1.6922	0.988	1.9037	0.889	2.1152	0.808	2.3467	0.741	2.5382	0.684	2.7498
4,200	1.782	1.3667	1.633	1.4692	1.508	1.5916	1.400	1.7140	1.225	1.9589	1.089	2.2037	0.980	2.4486	0.891	2.7335	0.817	2.9383	0.754	3.1832
4,400	1.946	1.5484	1.784	1.6892	1.655	1.8300	1.537	1.9707	1.344	2.2522	1.189	2.5338	1.071	2.8153	0.973	3.0969	0.892	3.3784	0.827	3.6599
4,800	2.327	2.0103	2.133	2.1930	1.969	2.3758	1.829	2.5585	1.600	2.9241	1.422	3.2896	1.280	3.6551	1.164	4.0206	1.067	4.3861	0.985	4.7516
5,200	2.731	2.5559	2.504	2.7882	2.311	3.0206	2.146	3.2530	1.871	3.7177	1.670	4.1824	1.502	4.6471	1.366	5.1118	1.252	5.5765	1.156	6.0412
5,600	3.168	3.1923	2.904	3.4825	2.680	3.7727	2.489	4.0629	2.178	4.6433	1.936	5.2237	1.742	5.8041	1.584	6.3845	1.452	6.9649	1.340	7.5453
6,000	3.636	3.9263	3.333	4.2833	3.077	4.6402	2.857	4.9959	2.500	5.7110	2.222	6.4249	2.000	7.1388	1.818	7.8527	1.667	8.5666	1.538	9.2804

Table No. 94.—Pressure and Horse-Power Required to Compensate for the Friction of Air Passing through Pipes.—Concluded.

Velocity of Air in Feet per Minute.	DIAMETER OF PIPE.																			
	28-inch.		30-inch.		32-inch.		36-inch.		40-inch.		44-inch.		48-inch.		52-inch.		56-inch.		60-inch.	
	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.	Loss of Press. in oz. per sq. in.	H. P. lost in Friction.
100	0.000	0.0000	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001	0.000	0.0001
200	0.002	0.0004	0.001	0.0004	0.001	0.0004	0.001	0.0005	0.001	0.0005	0.001	0.0006	0.001	0.0006	0.001	0.0007	0.001	0.0007	0.001	0.0008
300	0.004	0.0012	0.003	0.0014	0.003	0.0014	0.003	0.0016	0.002	0.0018	0.002	0.0020	0.002	0.0021	0.002	0.0023	0.002	0.0025	0.002	0.0027
400	0.006	0.0030	0.006	0.0032	0.006	0.0034	0.005	0.0038	0.004	0 0042	0.004	0.0047	0.004	0.0051	0.003	0.0055	0.003	0.0059	0.003	0.0063
500	0.010	0.0058	0.009	0.0062	0.009	0.0066	0.008	0.0074	0.007	0.0083	0.006	0.0091	0.006	0.0099	0.005	0.0107	0.005	0.0116	0.005	0.0124
600	0.014	0.0100	0.013	0.0107	0.012	0.0114	0.011	0.0128	0.010	0.0143	0.009	0.0157	0.008	0.0171	0.008	0.0186	0.007	0.0200	0.007	0.0214
700	0.019	0.0159	0.018	0.0170	0.017	0.0181	0.015	0.0204	0.014	0.0227	0.012	0.0249	0.011	0 0272	0.010	0.0295	0.010	0.0317	0.009	0.0340
800	0.025	0.0237	0.024	0.0254	0.022	0.0271	0.020	0.0305	0.018	0.0339	0.016	0.0372	0.015	0.0406	0.014	0.0440	0.013	0.0474	0.012	0.0508
900	0 032	0.0337	0.030	0.0361	0.028	0.0385	0.025	0.0434	0.022	0 0482	0.020	0.0530	0.019	0.0578	0.017	0.0626	0.016	0.0675	0.015	0.0723
1,000	0 040	0.0463	0.037	0.0496	0.035	0.0569	0.031	0.0595	0.028	0.0661	0.025	0.0727	0.023	0.0793	0.021	0.0859	0.020	0.0925	0.019	0.0991
1,100	0.048	0.0616	0.045	0.0660	0.042	0.0704	0.035	0.0792	0.034	0 0880	0.030	0.0968	0.028	0.1055	0.026	0.1144	0.024	0.1232	0.022	0.1320
1,200	0.057	0.0800	0.053	0.0857	0.050	0.0914	0.044	0.1024	0.040	0.1142	0.036	0.1256	0.033	0.1371	0.031	0.1485	0.029	0.1599	0.027	0.1713
1,300	0.067	0.1017	0.063	0.1089	0.058	0.1162	0.052	0.1307	0.047	0.1452	0.043	0.1597	0.039	0.1746	0.036	0.1888	0.034	0.2033	0.031	0.2178
1,400	0.078	0.1270	0.073	0.1360	0.068	0.1451	0.060	0.1632	0.054	0.1814	0.049	0.1995	0.045	0.2177	0.042	0.2360	0.039	0.2539	0.036	0.2721
1,500	0.089	0.1562	0.083	0.1673	0.078	0.1785	0.069	0.2008	0.062	0.2231	0.057	0.2454	0.052	0.2677	0.048	0.2900	0.045	0.3123	0.042	0.3346
1,600	0.102	0.1895	0 095	0.2031	0 089	0.2166	0.079	0 2437	0.071	0.2707	0.069	0.2938	0.059	0.3249	0.055	0.3520	0.051	0.3790	0.047	0.4061
1,700	0.115	0.2273	0.107	0.2436	0.103	0.2598	0.089	0.2923	0.080	0 3247	0.073	0.3572	0.067	0.3897	0.062	0.4220	0.057	0.4546	0.054	0.4871
1,800	0.129	0.2702	0.120	0.2891	0.112	0.3084	0.100	0.3469	0.090	0.3855	0.082	0.4240	0.075	0.4626	0.069	0.5011	0.064	0.5406	0.060	0.5782
1,900	0.143	0.3174	0 134	0.3401	0.125	0.3627	0.111	0 4080	0.103	0.4534	0 091	0.4987	0.084	0.5440	0.077	0.5890	0.072	0.6347	0.067	0.6800
2,000	0.159	0.3702	0 148	0.3966	0.139	0.4230	0.123	0.4759	0.111	0.5288	0.101	0.5817	0.093	0.6346	0.085	0.6874	0.079	0.7403	0.074	0.7932
2,200	0.192	0.4927	0.179	0.5279	0.168	0.5631	0.149	0.6334	0.134	0.7038	0.122	0.7742	0.112	0.8446	0.103	0.9150	0.096	0.9854	0.090	1.1607
2,400	0.239	0.6396	0.213	0.6855	0.200	0.6910	0.178	0.8224	0.160	0.9138	0.145	1.0051	0.133	1.0965	0.123	1.1879	0.119	1.2793	0.107	1.3706
2,600	0.268	0.8132	0.250	0.8714	0.234	0.9294	0 209	1.0456	0.188	1.1618	0.171	1.2779	0.156	1.3941	0.144	1.5103	0.134	1.6265	0.125	1.7427
2,800	0.311	1.0157	0.290	1.0884	0.272	1.1608	0.242	1.3059	0.218	1.4510	0.198	1.5961	0.181	1.7412	0.168	1.8864	0.156	2.0314	0.145	2.1765
3,000	0.357	1.2490	0.333	1.3388	0.312	1.4280	0.278	1.6062	0.250	1.7847	0.227	1.9632	0.208	2.1416	0.192	2.3201	0.179	2.4979	0.167	2.6771
3,200	0.406	1.5162	0.379	1.6246	0.356	1.7328	0.316	1.9494	0.284	2.1660	0.259	2.3826	0.237	2.5992	0.219	2.8157	0.203	3.0324	0.190	3.2489
3,400	0.459	1.8186	0.428	1.9487	0.413	2.0950	0.357	2.3382	0.321	2.5980	0.292	2.8578	0.268	3.1176	0.247	3.3764	0.229	3.6372	0.214	3.8970
3,600	0.514	2.1620	0.480	2.3132	0 450	2.4671	0.400	2.8156	0.360	3.0840	0.327	3.3924	0.300	3.7007	0.279	4.0091	0.257	4.3239	0.240	4.6259
3,800	0.573	2.5389	0.535	2.7206	0.501	2.9016	0.446	3.2683	0.401	3.6270	0.365	3.9897	0.334	4.3484	0.309	4.7519	0.287	5.0779	0.267	5.4406
4 000	0.635	2.9613	0.593	3.1731	0.556	3.3843	0.494	3.8074	0.444	4.2304	0.404	4.6934	0.370	5.0765	0.342	5.4995	0.317	5.9226	0.296	6.3456
4,200	0.700	3.4281	0.653	3.6729	0.612	3.9178	0.544	4.4075	0.490	4.8972	0.445	5.4669	0.408	5.8767	0.377	6.3664	0.350	6.8561	0.327	7.3458
4,400	0.768	3.9415	0.717	4.2234	0.672	4.5045	0.595	5.0676	0.535	5.6307	0.487	6.1937	0.446	6 7568	0.414	7.3199	0.384	7.8829	0.357	8.4460
4,800	0.914	5.1171	0.853	5.4836	0.800	5.8481	0.711	6.5791	0.640	7.3101	0.582	8.0411	0.533	8.7722	0.492	9.5032	0.457	10.2342	0.427	10.9642
5,200	1.073	6.5059	1.011	6.9713	0.935	7.4354	0.835	8 3648	0.751	9.2942	0.683	10.2236	0.626	11.1530	0.578	12.0824	0.537	13.0119	0.501	13.9413
5,600	1.244	8.1258	1.161	8.7070	1.089	9.2866	0.968	10.4474	0.871	11 6082	0.792	12.7690	0.726	13.9300	0.670	15.0907	0.622	16.2515	0.581	17.4123
6,000	1.429	9.9918	1.333	10.7100	1.250	11.4221	1.111	12.8498	1.000	14.2776	0.909	15.7054	0.833	17.1331	0.769	18.5609	0.714	19.9835	0.667	21.4164

allowance has been made for differences of temperature between the ends of the pipe. For any other length of pipe the losses will be directly proportional. From this table is evident the desirability, and in many cases the necessity, of making pipes of ample area. Suppose, for illustration, it is desired to move a given volume of air, such that, if passed through a 6-inch pipe, the velocity would be 4,000 feet per minute. The horse-power lost in friction is shown by the table to be 0.6346. If this same volume were to be passed through a 12-inch pipe which has four times the area, the velocity would require to be only one-fourth as great, or 1,000 feet per minute; and the loss in horse-power, per the table, would be only 0.0198, or one thirty-second of that expended in overcoming the resistance of a 6-inch pipe. The power in each instance being only that which is necessary to overcome the resistance, there is further required an expenditure of power to actually move the air. These two items of power, that necessary to overcome the pressure head, and that required to produce the velocity head, must be added together to determine the total amount of power necessary to move the air under the two conditions.

Tremendous losses are evident in the case of small pipes, but they become of relatively far less importance in the larger sizes. Thus the horse-power loss in the 6-inch pipe, when the velocity is 4,000 feet per minute, is 0.6346; and that in a 24-inch pipe, with the same velocity, is 2.5382. But the latter pipe having 16 times the area of the former, the actual loss per unit volume moved in the latter is $\frac{2.5382}{16} = 0.1586$, or one-fourth of that lost in the movement of the same volume of air at the same velocity in the 6-inch pipe.

The conclusion to be drawn from this table is that excessively small pipes and high velocities should, so far as possible, be avoided; but that, after a reasonable size is reached, or the velocity is brought down to a moderate rate, a change in either size or velocity will have only a relatively small effect upon the loss in pressure and power. It is, therefore, evident that if the attempt to reduce the losses is carried to an extreme, other factors, such as the cost of the pipe and the space occupied, may turn such apparent saving into a practical loss from a commercial standpoint.

Measurement of Draft. — Draft is usually measured by the difference in level of a liquid in the arms of a tube of U form having one end open to the atmosphere and the other connected with the enclosed space within which a different pressure exists. The preponderance of pressure in one arm forces the liquid downward and causes a corresponding rise in the other. The difference in level represents the height of a column of the liquid which will be sustained by the excess of pressure. For large pressure differences, mercury is used because of

Table No. 95.—Pressures in Ounces per Square Inch Corresponding to Various Heads of Water in Inches.

Head in Inches.	Decimal Parts of an Inch.									
	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0		0.06	0.12	0.17	0.23	0.29	0.35	0.40	0.46	0.52
1	0.58	0.63	0.69	0.75	0.81	0.87	0.93	0.98	1.04	1.09
2	1.16	1.21	1.27	1.33	1.39	1.44	1.50	1.56	1.62	1.67
3	1.73	1.79	1.85	1.91	1.96	2.02	2.08	2.14	2.19	2.25
4	2.31	2.37	2.42	2.48	2.54	2.60	2.66	2.72	2.77	2.83
5	2.89	2.94	3.00	3.06	3.12	3.18	3.24	3.29	3.35	3.41
6	3.47	3.52	3.58	3.64	3.70	3.75	3.81	3.87	3.92	3.98
7	4.04	4.10	4.16	4.22	4.28	4.33	4.39	4.45	4.50	4.56
8	4.62	4.67	4.73	4.79	4.85	4.91	4.97	5.03	5.08	5.14
9	5.20	5.26	5.31	5.37	5.42	5.48	5.54	5.60	5.66	5.72

the relatively small range which, owing to its great density, is required to show great differences in pressure. Water, however, is ordinarily employed in the measurement of draft in connection with steam boilers. One cubic foot (1,728 cubic inches) of water at 50°, the temperature at which the preceding calcula-

Table No. 96.—Height of Water Column in Inches Corresponding to Various Pressures in Ounces per Square Inch.

Pressure in Ounces per Square Inch.	Decimal Parts of an Ounce.									
	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0		0.17	0.35	0.52	0.69	0.87	1.04	1.21	1.38	1.56
1	1.73	1.90	2.08	2.25	2.42	2.60	2.77	2.94	3.11	3.29
2	3.46	3.63	3.81	3.98	4.15	4.33	4.50	4.67	4.84	5.01
3	5.19	5.36	5.54	5.71	5.88	6.06	6.23	6.40	6.57	6.75
4	6.92	7.09	7.27	7.44	7.61	7.79	7.96	8.13	8.30	8.48
5	8.65	8.82	9.00	9.17	9.34	9.52	9.69	9.86	10.03	10.21
6	10.38	10.55	10.73	10.90	11.07	11.26	11.43	11.60	11.77	11.95
7	12.11	12.28	12.46	12.63	12.80	12.97	13.15	13.32	13.49	13.67
8	13.84	14.01	14.19	14.36	14.53	14.71	14.88	15.05	15.22	15.40
9	15.57	15.74	15.92	16.09	16.26	16.45	16.62	16.79	16.96	17.14

tions have been made, weighs 62.409 pounds; therefore a column of water of this temperature 1,728 inches high and 1 square inch cross-sectional area would exert a pressure of 62.409 pounds per square inch, and a pressure of 1 pound per square inch would be exerted by a column $\frac{1{,}728}{62.409} = 27.7$ inches high. From this it is readily deduced that an ounce pressure per square inch is produced by a water column 1.73 inches high, and that 1 inch head of water is equivalent to a pressure of 0.578 ounces per square inch. Table No. 95 serves to show these relations for different heights of water column. Table No. 96 indicates the height of water column corresponding to any given pressure in ounces per square inch.

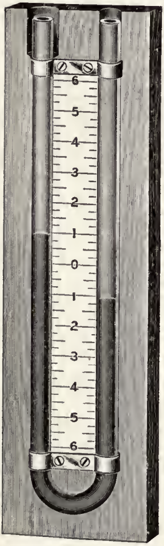

FIG. 5.
WATER GAUGE FOR LOW PRESSURES.

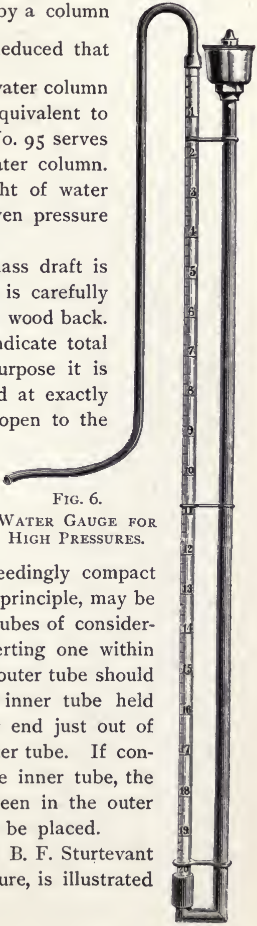

FIG. 6.
WATER GAUGE FOR HIGH PRESSURES.

The simplest form of U-tube glass draft is shown in Fig. 5. The glass tube is carefully fitted and securely attached to the wood back. The scale is laid out so as to indicate total differences in level, for which purpose it is necessary to have the water stand at exactly zero in both tubes when they are open to the atmosphere. For reading other pressures, a rubber tube may be slipped over the end of the left-hand tube, where the wood back is cut away.

A somewhat unique and exceedingly compact form of gauge, acting upon this principle, may be constructed by making the glass tubes of considerably different diameters and inserting one within the other. The lower end of the outer tube should be permanently closed, and the inner tube held rigidly in position, with its lower end just out of contact with the bottom of the outer tube. If connection be made to the top of the inner tube, the change in level may be clearly seen in the outer tube, upon the surface of which the graduations may be placed.

A substantial form of gauge, manufactured by the B. F. Sturtevant Co., and capable of measuring up to 20 ounces pressure, is illustrated in Fig. 6.

One arm of the U tube is made of brass tubing, and provided with a cup at the top of such size that practically constant level is maintained therein under the different conditions of pressure. The scale is constructed with this level as its zero mark.

Gauges of the form and construction just described serve all ordinary purposes, but lack the refinement necessary to the determination of very small differences of pressure. For this purpose special devices are necessary. One of these is shown in Fig. 7. This embodies the principle of the U tube; but the level of the water is very closely ascertained by the aid of a hook gauge. As is evident in the illustration, this instrument consists, in effect, of two tubes of relatively large diameter made in one casting. The tubes communicate near the bottom through an opening, whereby a constant level of the water within them may be maintained so long as they are both exposed to the same pressure. The hook having been set so that its point is just at the surface of the water, and the micrometer reading having been taken, connection may be made to the air cock at the left, and the hook again adjusted at the new level. The micrometer screw, with its graduated head, makes the reading of pressure differences to $\frac{1}{100}$ of an inch a comparatively simple matter.

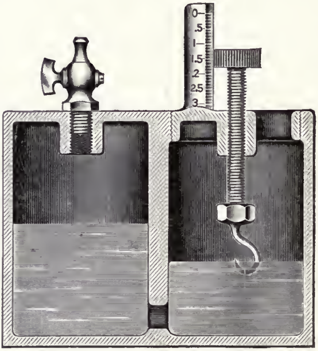

Fig. 7.
Hook Draft Gauge.

Evidently, such a gauge gives only relative readings, but may be so graduated as to indicate the total difference in water levels. These two readings, the difference between which gives the pressure in inches of water, render it unnecessary to bring the water to any stated level under atmospheric pressure.

An extremely delicate but easily read draft guage is that described by Weisbach, under the name of the Wollaston Anemometer, and perfected by Messrs. J. C. Hoadley and F. H. Prentiss.[1] As constructed and used by them, it consists of two glass tubes (Fig. 8) about 30 inches long and about 0.4 inches diameter inside, connected at each end, by means of stuffing-boxes, to suitable tubular attachments, through which they are secured to a backing of wood. A stop-cock in each of these attachments serves to establish or shut off communication between the glass tubes. Connecting with the top of each is a brass drum 4.25 inches in diameter, with heads of glass. Each drum is provided with a nipple and stop-cock for connection by tube to any desired space. Two sliding scales are provided between the glass tubes, to measure, one the depressions, and the other the elevations, of the liquid filling the lower half of the tubes.

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

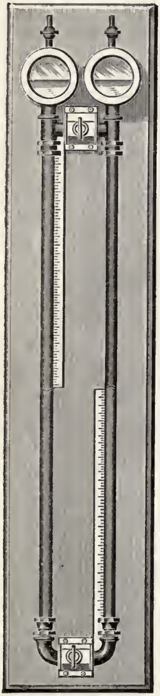

Fig. 8.
Modified Wollaston Anemometer.

The lower stop being open, the two tubes are filled up to about the middle of their heights with a mixture of alcohol and water. The lower stop-cock is then closed, the upper one opened, and crude olive oil is carefully poured in until it fills the first tube up to the upper cross tube, whence it flows into the second tube, and so finally fills both tubes and rises to the middle of both drums.

The oil forms, with the water-and-alcohol mixture, a very fine meniscus, which is readily discernible because of the contrast in colors. The specific gravity of oil should differ from that of the mixture by at least 1 per cent to avoid the tendency to mix. For most purposes, a difference of 2 per cent in the specific gravity will give sufficient sensitiveness — fifty times as much as a water column.

The method of using this instrument is as follows: The stop-cocks between the tubes and those on the drums are opened and the liquids allowed to come to a level. If, now, one of the drums be connected with a suction fan or chimney, the diminished pressure will cause the oil to flow up into that drum. The surface of the oil in the drum is about 100 times as large as the inside cross-section of the glass tubes, and in the same proportion will the rise of the lower liquid on the one side, and its depression on the other, exceed the corresponding rise and depression of the upper surface of the oil.

If, now, when equilibrium has been restored, the lower stop-cock be closed, the upper one opened, and the connection with the fan or chimney severed, the lower liquid will be kept immovable, while the oil will flow through the upper cross-tube, and come again to common level in the two drums. On connecting again with the flue or chimney, and again closing the upper stop-cock and opening the lower one, a diminished repetition of the former action will take place; the lower liquid will rise a little in one tube, and fall a little in the other, and the surface level of the oil in the two drums will become slightly unequal. A few repetitions of this process will bring the difference in level of the lower liquid in the two tubes to represent the entire difference in pressure on the surface of the oil in the two drums. That is, a certain known height of column, filled, in one tube with a mixture of alcohol and water, with the fan or chimney pressure on its surface, is just balanced by an equal height of column filled with olive oil, with the pressure of the atmosphere on its surface. If the specific gravity of the oil be 0.916, and that of the mixture of water and alcohol be 0.926, their difference is only 1 per cent. As specific gravity is referred to water as unity, the differential column existing where the liquids differ in specific gravity by only 1 per cent represents the same effect as that of a water column $\frac{1}{100}$ part as high or a mercury column $\frac{1}{1,360}$ part as high. This instrument, with the respective specific gravities of 0.976 and 0.937, was sensitive enough, as applied by Hoadley, to show plainly the reduction of chimney draft caused by opening a sliding register in the fire door for the admission of air above the fire, although the aggregate area of the opening was only six square inches.

All of the gauges which have been described are designed only for independent observations, so that an approach to a continuous record can only be secured by a multitude of readings taken at very short intervals. The impracticability of such a method points to the advantages of an instrument which by its own operation records the changes in the intensity of the draft. Such is the Mechanical Draft Recorder shown in Fig. 9. The instrument, which is specially constructed for this Company by the Crosby Steam Gage and Valve Company, consists of two essential parts. First, the small cylinder, in which operates a practically frictionless piston under the influence of the draft. The motion of this piston is like that of a steam-engine indicator, multiplied by the attached arm, which carries at its end a reservoir containing ink. The second essential portion is the dial or chart, which is usually graduated so as to indicate the pressure or vacuum in inches of water. This chart, which is of paper, is held in place upon a circular

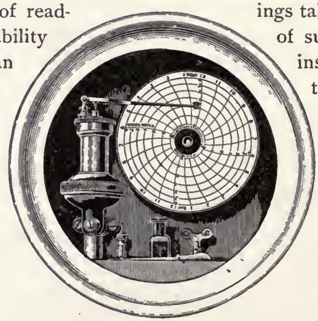

Fig. 9.
Mechanical Draft Recorder.

plate which is caused to revolve by a system of clockwork. The point of the ink reservoir, being kept elastically in contact with the revolving dial, continuously records all variations in the draft.

In measuring the pressure exerted by moving air, both the velocity head and the pressure head have to be taken into account. To separate these two factors of the total head, a form of Pitot's tube may be employed, as illustrated in Fig. 10, where it is applied in connection with a pipe, through the side of which it is inserted. The tube A is open at the end and connects by rubber tubing with one arm of an ordinary U-tube water gauge. The other tube B is closed upon the end, but has in its opposite sides two small holes, and is connected to the other arm of the gauge. Tube A receives the full effect of the current of moving air, and thus tends to indicate upon the gauge the total head, including both the velocity head and the pressure head. But the influence of the velocity is practically removed from B, which, therefore, receives only the pressure due to the pressure head. As this tube is connected to the other arm of the gauge, the pressure thus indicated is only that due to the velocity head; for, both arms being subject to the pressure head, these pressures are balanced.

At high pressures even this device is not altogether reliable, for the air moving by the openings in tube B has an aspirating influence which may tend to produce a partial vacuum in this tube. It is, therefore, necessary, before making final measurements, to determine by independent readings whether this is the case, and to what extent.

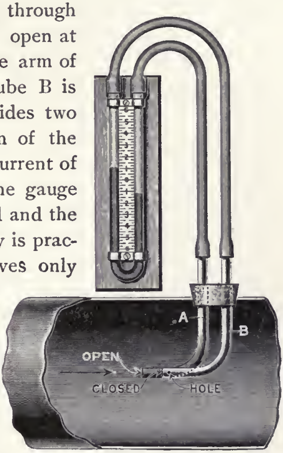

FIG. 10. PITOT'S TUBE.

Conditions of Boiler Draft. — In boiler practice the force of the draft must be expended in two ways. First, a portion of it is necessary to overcome the resistances of the grate and the fuel upon it, of the combustion chamber, flues or tubes, and uptake, and of the means of connection to the source of draft, be it fan or chimney. Within the chimney or fan certain other resistances must also be overcome. Second, the draft must in addition be sufficient to impart the necessary velocity to the requisite amount of air for the direct purposes of combustion. As has already been shown, the velocity thus produced varies directly as the square root of the intensity of the draft, and consequently the volume at constant temperature likewise varies in the same ratio. The force expended in overcoming the resistances is directly proportional to the pressure,

—that is, to the square of the velocity,—while the work done in moving the air, being the resultant of a given pressure exerted through a given distance, which is measured by the velocity, becomes proportional to the product of these factors; namely, to the cube of the velocity. Therefore, if under stated conditions the resistance be increased, as by a thicker fire or finer coal, the intensity of draft required for overcoming this resistance must also be increased. If, however, the prior conditions were such that the maximum intensity of draft attainable was already devoted to the requirements of combustion, any demand for increased draft to overcome the greater resistance could only be met by reducing the amount of that portion previously devoted to the creation of velocity. But a slight reduction in the velocity considerably decreases the expenditure necessary for overcoming the resistance. The ultimate result, however, is that the expenditure for overcoming resistance is increased, while the velocity and consequent volume of air are decreased. But both of these changes are proportionately less than would at first appear. The reason is evident in the fact that, whereas the resistances which are overcome are directly proportional to the pressure or draft exerted, the velocity is decreased only in proportion to the square root of the draft, which is thus diverted from its former service of producing velocity to that of overcoming the added resistance.

In all boiler practice the most important of the resistances to be overcome are those of the tubes and of the grate with the fuel upon it, while the expenditure for the production of velocity is comparatively small. A careful study of these resistances is, therefore, of importance in a thorough consideration of the practical conditions of draft production.

Prof. Gale[1] found, in the case of a stationary boiler furnace of ordinary construction, the following pressures in pounds per square foot:—

Required to produce entrance velocity (3.6 feet per second) .	0.013
Required to overcome resistance of fire grate	0.91
Required to overcome resistance of combustion chamber and boiler tubes	1.23
Required to overcome resistance in horizontal flue	0.06
Required to produce discharge velocity (11.2 feet per second),	0.085
Total effective draft pressure	2.298
Back pressure due to friction in stack	0.19
Total static pressure produced by chimney	2.488

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[1] Theory and Design of Chimneys. Horace B. Gale. Transactions American Society of Mechanical Engineers, Vol. XI.

This total static pressure, which is given in pounds per square foot, is equivalent to 0.28 ounces per square inch, or 0.48 inches of water.

It is evident from this table that the greater part of the draft pressure is, as already stated, necessary to overcome the resistances presented by the fuel and the boiler tubes. In fact, Rankine asserts that the throttling action at the grate is ordinarily sufficient to cause a loss of head equal to about three-quarters of the whole draft of a chimney. As about 75 per cent of this remaining fourth is necessary to balance the frictional resistances of the flues and the chimney, there remains only about one-sixteenth of the total head for the production of velocity. Prof. H. B. Gale[1] cites an instance in which he found that about 60 per cent of the entire head was lost by throttling at the grate, and only about 4 per cent of the total head was actually expended in accelerating the gases. In this case the height of the chimney was 92 feet, and the temperature of the gases 609°. If the whole head had been employed in producing velocity of the gases, their velocity, as calculated by Prof. Gale, would have been about 78 feet per second. In reality the mean observed velocity was only 16 feet per second.

Although the ultimate object of any means of draft production must necessarily be to create draft or velocity sufficient to provide the required amount of air and to carry off the gases, yet this portion of its work is almost infinitesimal as compared with the demand made for sufficient pressure to overcome the resistance of the fuel and the boiler. In other words, the ability to create sufficient pressure difference is the primary requisite to burning a given quantity of fuel, rather than the ability to move a certain amount of air. Draft-producing apparatus is not, therefore, to be based merely upon the total number of cubic feet to be moved per hour, as determined by multiplying the coal consumption by the allowance of air per pound of coal. If this were the case, a low chimney, or a large slow-running fan, would meet the requirements. In reality, the relatively immense resistances of fuel and boiler demand that the chimney or fan shall first be designed to create sufficient intensity of draft to overcome these resistances and to create the requisite velocity. This velocity must be such that, if multiplied by the full area at which it is measured, the product will equal the volume of air necessary for the combustion of the stated amount of fuel. The height of chimney or the diameter and speed of fan necessary to create the draft thus shown to be required having been determined, it is only necessary to make the capacity such as to accommodate the given volume of air.

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[1] Transactions American Society of Mechanical Engineers, Vol. XI., p. 777.

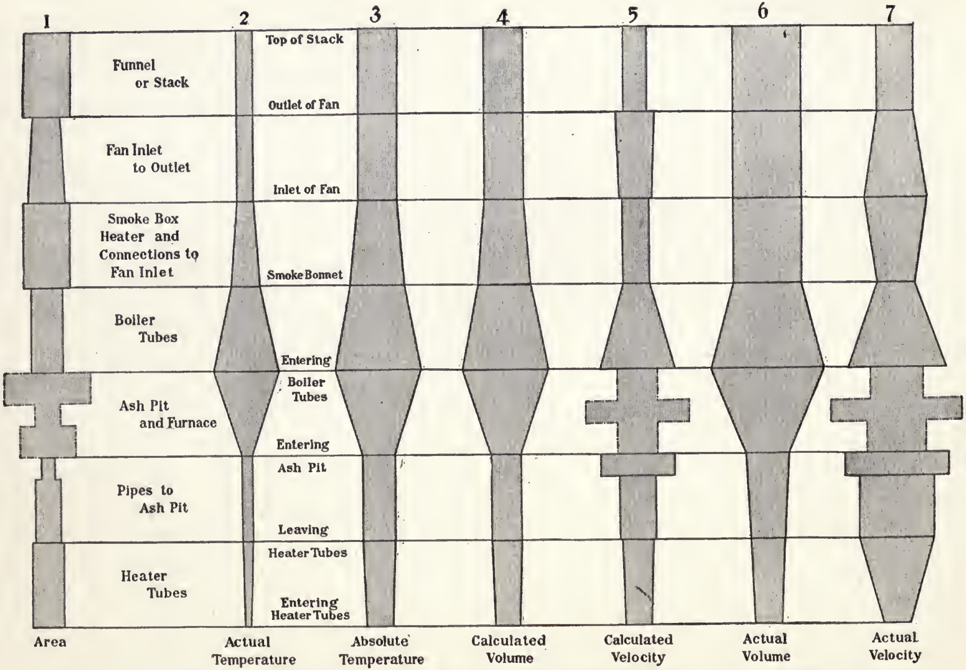

FIG. 11. RELATION OF AREA FOR PASSAGE, TEMPERATURE, VOLUME AND VELOCITY OF GASES IN BOILER OF STEAMSHIP "BERLIN" OF INTERNATIONAL NAVIGATION COMPANY.

Of course the velocity with which the air and gases pass from the ashpit to the uptake changes greatly as they progress, owing to the variations in area and temperature. All of these changes play their part in affecting the draft required to secure the desired results. The conditions in practice are very clearly shown by Fig. 11, which represents in graphical form the relative areas, temperatures, volumes and velocities, as determined by one of B. F. Sturtevant Co.'s expert engineers in the case of one of the boilers of the International Navigation Company's steamship Berlin. This boiler was one of four connecting with the same stack. It was provided with the Ellis & Eaves system of preheating the air before admission to the ashpit, and the necessary draft was produced by four special Sturtevant steam fans, through which the gases were drawn and thence discharged into the funnel. The relative actual observed temperatures are indicated in diagram 2, together with a statement of the portion of the passage to which they pertain. The first diagram graphically represents the area existing at the various points, and the third indicates the corresponding absolute temperatures. Taking the volume of air entering the heater tubes as unity, the fourth diagram has been constructed, showing the relative volumes to which the air would in each case be expanded by the existing temperature; while the fifth diagram, also calculated, indicates the velocities required to move the given volume through the given area. Both of these diagrams are purely theoretical, for they make no allowance for leakage or for increased volume due to the accession of the products of combustion. Diagrams 6 and 7, however, present the conditions as actually observed, and when compared with diagrams 4 and 5 clearly show the effect of combustion and of the leakage, which, owing to existing conditions, was large. The area through the fuel and the factors depending upon it are only suggested by the dash and dot lines.

Relation of Draft and Rate of Combustion. — It has been the usual practice, in the determination of the draft in connection with a given boiler, to ascertain only the total draft, as shown by the application of a gauge to the uptake, chimney or fan. From such readings it is manifestly unfair to draw conclusions as to the amount or intensity of draft to secure a given rate of combustion. For, first, the boiler resistances will, with the same boiler, remain practically constant, while the character of the fuel and the fire may vary greatly; and, second, with the same conditions as to fuel and combustion, a change in the type of boiler may increase or decrease the resistance due thereto. It is, therefore, evident that for comparison of combustion rates, the draft which should be determined is that relating solely to the supply of air to, and the overcoming of the resistances in, the fire. This draft is obviously the difference between the over- and under-grate pressures.

Although it has already been shown that the efficiency of combustion increases and the required air supply decreases as the combustion rate rises, for the sake of simplicity in the matter of comparison the required air supply per pound of coal may here be taken as constant for all rates of combustion. The volume of air (for equal temperature) then becomes an index of its velocity, and varies as the square root of the effective pressure or draft. Conversely, the required draft will vary as the square of the rate of combustion. Of course, for the purpose of properly proportioning draft-producing apparatus, the resistances of all parts of the boiler, including the fuel upon the grate, should be ascertained for all ordinary coals, types of boiler and conditions of draft.

The difficulties in the way of obtaining such knowledge are evident. It is not easy to secure identical conditions of boiler and draft when testing different coals, and even with the same coal there may be variations in its size, in the manner in which it is fired and in which the grates are kept clear of ashes, which very seriously affect the results. As indicative of the variation in draft pressure required for different kinds of coal, rates and stages of combustion, the results in Table No. 97 are presented. These are from a test[1] of a Coxe stoker applied to Babcock & Wilcox boilers. This stoker, which is of the travelling chain-grate type, with the fire upon its upper surface, is provided with four blast compartments under the fire. To each of these air is admitted in the desired proportion from a supply furnished by a Sturtevant fan.

Table No. 97.—Draft Conditions with Coxe Stoker.

ITEMS.		Size of Coal.		
		Buckwheat.	Buckwheat.	Rice.
Pressure in igniting compartment,	inches of water	0.14	0.25	0.44
Pressure in burning compartment,	" "	0.31	0.56	0.89
Pressure in burning-down compartment,	" "	0.24	0.49	0.73
Pressure in burning-out compartment,	" "	0.17	0.42	0.67
Pressure of blast of air, average,	" "	0.24	0.43	0.68
Vacuum in furnace,	" "	0.10	0.15	0.24
Total furnace draft,	" "	0.34	0.58	0.92
Vacuum in stack flue,	" "	0.13	0.40	0.58
Total draft,	" "	0.37	0.83	1.26
Pounds of dry coal per hour per square foot of grate		19.8	32.9	28.0

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[1] Experiments with Automatic Mechanical Stokers. J. M. Whitham. Transactions of American Society of Mechanical Engineers, Vol. XVII.

Mr. Whitham, in the same paper, also presented a table given here as Table No. 98, showing the relation of size of coal to results obtained with a Wilkinson stoker, from which the increased draft required by small sizes of coal is made evident. It is here that mechanical draft becomes most beneficial in making possible the combustion of low-grade (because finely divided) fuel.

Table No. 98. — Relative Rates of Combustion of Small Sizes of Anthracite Coal.

Grade of Coal.	Size of Coal (Round Holes, Punched Plates).	Relative Rates of Combustion for Same Draft.
Pea.	Through ⅞ inch and over 9-16 inch	100
Buckwheat.	Through 9-16 inch and over ⅜ inch	85
Rice.	Through ⅜ inch and over 3-16 inch	70

Present methods of boiler testing lack the refinements in ascertaining the draft pressure at different points which are necessary to an intelligent comparison of results. This fact is emphasized by the results given in Table No. 99,[1] which presents the observed conditions of draft and mean indicated horse-power per square foot of grate, in the trial tests of a number of steam vessels almost identical in their steam-power equipment. Draft was in each case produced by fans discharging into a closed fire room. The fact that under similar conditions the fire-room pressures varied from 1.02 to 2.01 inches is, at least, sufficient to emphasize the necessity of readings which indicate the effective pressure differences between ashpit and furnace chamber.

Table No. 99. — Relation of Air Pressure and Indicated Horse-Power per Square Foot of Grate.

Name of Ship.	Mean Air Pressure in Fire Rooms. Inches.	Mean I. H. P. per Square Foot of Grate Surface.
Orlando	1.02	16.17
Undaunted	1.87	16.17
Australia	1.77	17.75
Galatea	1.14	18.50
Narcissus	1.02	16.17
Immortalitè	2.01	17.93

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[1] Artificial Draft and its Effects on Boiler Construction. E. Lechner. Translated by Asst. Engr. Emil Theiss, U. S. Navy. Journal of American Society of Naval Engineers, Aug., 1891.

Under practically identical conditions of boiler, coal and firing, there is still opportunity, particularly with mechanical draft, for divergence from any established relation, owing to a probable decrease in air volume per pound of coal as the rate of combustion increases, and to a coincident increase in the thickness of the fire. These tend, however, to counteract each other. A greater depth of fuel on the grates naturally indicates a proportionately greater resistance. But the rate of combustion always increases at a more rapid rate than the thickness of the fire; therefore, the pressure under which the greater volume of air would be supplied would also increase more rapidly than the thickness of the fire, and hence more readily tend to overcome this resistance.

The combined effect of all the factors appears to be to bring the rate of combustion to substantially the theoretical basis first presented; viz., proportional to the square root of the effective pressure. Thus, for instance, two similar tests upon the same boiler in the works of the B. F. Sturtevant Co. gave results presented in Table No. 100. The square roots of the effective pressures in the two cases are, respectively, 0.705 and 0.542, which are in the ratio of 1 to 0.769, while the rates of combustion are in the ratio of 1 to 0.767.

Table No. 100. — Relation of Draft and Rate of Combustion in Boiler at B. F. Sturtevant Co.'s.

Designation of Test.	Vacuum in Furnace. Inches.	Vacuum in Ashpit. Inches.	Effective Pressure to Produce Combustion. Inches.	Coal Burned per Hour per Sq. Ft. of Grate. Pounds.
A	0.640	0.144	0.496	21.45
B	0.372	0.078	0.294	16.45

The results of a series of tests of the locomotive boiler of a torpedo boat, as presented by E. Lechner[1] in a discussion of this subject, are given in Table No. 101. Tests numbered 1, 2, 3 and 4 were conducted upon the regular grate under artificial draft produced by a centrifugal fan in a closed fire room, while in tests numbered 5, 6 and 7 the grate was reduced to one-half the area. In the first series of tests the air supply per pound of coal was reasonable in quantity, but practically constant for different rates of combustion; while in the second series, owing to the existing circumstances, it was large in volume but decreased as the rate of combustion increased.

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[1] Artificial Draft and its Effects on Boiler Construction. E. Lechner. Translated by Assistant Engineer Emil Theiss, U. S. Navy. Journal of American Society of Naval Engineers, August, 1891.

Table No. 101.—Results of Tests of Locomotive Boiler on Torpedo Boat with Different Rates of Combustion.

Number of Test.	Grate Surface. Square Feet.	Heating Surface. Square Feet.	Ratio of Heating to Grate Surface.	Gauge Pressure. Pounds per Square Inch.	Air Pressure in Inches of Water.		
					Fire Room.	Furnace.	Chimney.
1	20.44	882.32	43.1	113.6	1.97	1.57	0.39
2	——	——	——	——	2.95	2.36	0.59
3	——	——	——	——	3.94	3.15	0.91
4	——	——	——	——	5.90	4.53	1.18
5	10.22	882.32	86.2	113.6	2.95	2.17	0.39
6	——	——	——	——	5.90	3.94	1.00
7	——	——	——	——	6.80	4.53	1.18

Number of Test.	Coal per Hour per Square Foot of Grate. Pounds.	Water Evaporated per Hour from 86°. Pounds.	Water Evaporated per Pound of Coal. Pounds.	Temperature of Chimney Gases. Degrees.	Air Supplied per Minute. Cubic Feet.	Air per Pound of Coal. Cubic Feet.	Mean Thickness of Fuel on Grates Inches.
1	53.6	437.8	8.16	518°	4,112	225.2	12.60
2	63.4	517.5	8.17	608	5,365	248.4	13.40
3	76.4	583.6	7.64	626	6,287	241.6	14.17
4	93.4	649.8	7.00	716	7,092	222.9	15.35
5	66.7	528.6	7.93	554	5,221	459.6	9.84
6	101.4	710.0	7.00	608	6,831	395.5	11.82
7	113.0	770.2	6.80	698	7,092	368.4	13.78

The relation existing between the rate of combustion and the square root of the corresponding pressure difference is presented in Table No. 102. A compar-

Table No. 102.—Relation of Square Root of Pressure Difference to Rate of Combustion.

Number of Test.	Difference of Pressure between Fire Room and Furnace. Inches.	Square Root of Difference of Pressure between Fire Room and Furnace.	Ratio of Square Roots of Difference of Pressure, referred to Test No. 1 as Unity.	Ratio of Square Roots of Difference of Pressure, referred to Test No. 5 as Unity.	Ratio of Corresponding Rate of Combustion, referred to Test No. 1 as Unity.	Ratio of Corresponding Rate of Combustion, referred to Test No. 5 as Unity.
1	0.40	0.632	1.00		1.00	
2	0.59	0.768	1.21		1.18	
3	0.79	0.888	1.40		1.42	
4	1.37	1.170	1.85		1.74	
5	0.78	0.883		1.00		1.00
6	1.96	1.400		1.59		1.52
7	2.27	1.507		1.71		1.70

ison of columns 4 and 5, or of columns 6 and 7, indicates that the rate of combustion advances in approximately the same proportion as the square root of the pressure difference, taking each series of tests by itself. An attempt to compare the second series with the first results in a series of values which do not continue the ratio of the first series, but which, if multiplied by a constant quantity, may be brought into accord therewith. The approximate relation between rate and pressure is thus indicated, as well as the fact that a complete formula, which shall enable one to determine the pressure difference required for a given rate of combustion for any kind of fuel, must include a constant which shall apply only under the given conditions, and whose various values for all conditions can only be determined by experiment.

Such a formula would naturally take the form —

$$m = c\sqrt{p_1 - p_2}$$

In which w = pounds of coal burned.
p_1 = pressure in ashpit.
p_2 = pressure in furnace chamber.
c = constant dependent on type of boiler, kind of fuel, depth of fire, etc.

But until the value of c is determined for all conditions, the formula must be limited in its application. In a word, the relation between the *effective* pressures and different rates of combustion under the same conditions is substantially established. But the exact draft or pressure difference required to maintain a given rate of combustion of a specific kind of coal in a particular type of boiler can only be approximated in the present state of knowledge.

The relation between the *total* draft and the rate of combustion, which is that usually indicated in the results of an ordinary test, is practically all that is at present available. But this relation is not of necessity the same as that between the *effective* pressure and the rate. In fact, the constant character of the resistances of the boiler proper and the variableness in those of the fuel with different rates cause this relation to depart somewhat from that holding in the case of the effective pressures.

This is clearly indicated in the results of careful tests of Mr. J. M. Whitham,[1] in which the draft was taken at various points in the passage of the air and gases through the furnace and boiler when coal was being burned at different rates of combustion. The boiler was of the horizontal return tubular type, 60 inches diameter by 20 feet long, with 44 4-inch tubes. The grate was stationary, 5 feet by 5 feet 4 inches, with 46 per cent air space, the ratio of heating surface to grate surface being 42.6. In certain tests the tubes were fitted with retarders made of strips of No. 10 iron 20 feet long, with a pitch of ten feet. Cambria Company coal (run-of-mine, bituminous) was used. The recorded drafts and rates of combustion are given in Table No. 103. The results are also graphically presented in Fig. 12, in which "fair" lines are drawn through the various points plotted. There is also added a line based upon the relation already stated,— that the rate of combustion should vary as the square root of the effective draft. Taking the furnace draft as representative of this effective draft, and starting at a rate of 12 pounds as unity, the corresponding theoretical drafts have been calculated and plotted for other rates. These correspond with those for the furnace draft, except at very low rates of combustion, but diverge decidedly from the lines representative of other elements of the total draft.

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[1] The Effect of Retarders in Fire Tubes of Steam Boilers. J. M. Whitham. Transactions American Society of Mechanical Engineers, Vol. XVII.

Table No. 103. — Relation of Draft and Rate of Combustion.

Pounds of Dry Coal burned per Hour per Square Foot of Grate.	Furnace Draft.	Resistance, in Inches of Water, Due to			Total Draft in Inches of Water.			
		Pass under Boiler and through Tubes. No Retarders.	Retarders in Tubes.	Pass over top of Boiler.	No Retarders. No Return Pass.	With Retarders. No Return Pass.	Top Pass. No Retarders.	With Top Pass and Retarders.
5	0.04	0.04	0.00	0.04	0.08	0.08	0.12	0.12
8	0.11	0.05	0.02	0.04	0.16	0.18	0.20	0.22
10	0.13	0.07	0.03	0.05	0.20	0.23	0.25	0.28
12	0.17	0.07	0.04	0.05	0.24	0.28	0.29	0.33
14	0.19	0.10	0.03	0.05	0.29	0.32	0.34	0.37
15	0.20	0.11	0.03	0.05	0.31	0.34	0.36	0.39
16	0.21	0.12	0.03	0.05	0.33	0.36	0.38	0.41
18	0.23	0.13	0.06	0.05	0.36	0.42	0.42	0.48
20	0.24	0.16	0.08	0.06	0.40	0.48	0.46	0.54
22	0.26	0.18	0.12	0.06	0.44	0.56	0.50	0.62
25	0.27	0.22	0.19	0.06	0.49	0.68	0.55	0.74
28	0.29	0.24	0.27	0.07	0.53	0.80	0.60	0.87
30	0.30	0.27	0.31	0.07	0.57	0.88	0.64	0.95
34	0.32	0.31	0.38	0.08	0.63	1.01	0.71	1.09
36	0.33	0.34	0.40	0.08	0.67	1.07	0.75	1.15
40	0.36	0.38	0.46	0.08	0.74	1.20	0.82	1.28

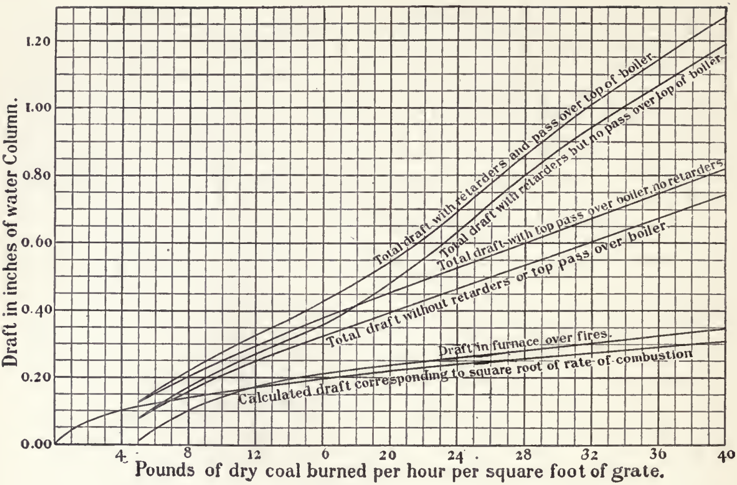

FIG. 12. RELATION OF DRAFT AND RATE OF COMBUSTION.

The total draft required for the efficient combustion of various kinds of fuels, as given by Hutton,[1] is presented in Table No. 104. Of course this must be considered as applying only under ordinary conditions, and as approximate at

Table No. 104.—Total Draft Required for Efficient Combustion of Different Kinds of Fuel.

KIND OF FUEL.	Total Draft in Inches of Water.	KIND OF FUEL.	Total Draft in Inches of Water.
Straw	0.20	Slack, very small . . .	0.7 to 1.1
Wood	0.30	Coal Dust	0.8 to 1.1
Sawdust	0.35	Semi-anthracite coal . .	0.9 to 1.2
Peat, light	0.40	Mixture of Breeze and Slack,	1.0 to 1.3
Peat, heavy	0.50	Anthracite, round . .	1.2 to 1.4
Sawdust mixed with small coal,	0.60	Mixture of Breeze and Coal Dust,	1.2 to 1.5
Steam coal, round . .	0.4 to 0.7	Anthracite Slack . . .	1.3 to 1.8
Slack, ordinary . . .	0.6 to 0.9		

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[1] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

the best. Hutton also gives tables of draft pressures and rates of combustion for different chimney heights, from which Table No. 105 has been compiled. He does not, however, specify the exact conditions for which they are calculated, but the figures are approximately true for medium-size anthracite coal and stationary boilers.

Table No. 105. — Rate of Combustion for Different Total Draft Pressures.

Height of Chimney above Grate, in Feet.	Total Draft Pressure in Inches of Water.	Rate of Combustion per Hour per Square Foot of Grate, in Pounds.	Height of Chimney above Grate, in Feet.	Total Draft Pressure in Inches of Water.	Rate of Combustion per Hour per Square Foot of Grate, in Pounds.
25	0.182	10	130	0.948	30
50	0.364	16	140	1.029	34
60	0.437	17	150	1.095	40
70	0.512	18	180	1.313	50
80	0.583	19	200	1.459	60
90	0.657	20	225	1.641	70
100	0.729	22	250	1.825	80
110	0.802	24	300	2.189	90
120	0.875	27	400	2.553	112

Leakage of Air. — A fruitful source of poor draft and decreased efficiency lies in the leakage of air through boiler settings. The extent of such infiltration is frequently surprising, being often so great that the flame of a match is drawn to and into the interstices of an 8-inch brick wall, not alone at fine visible cracks, but at mortar joints apparently sound. Evidently, the result of such leakage with suction draft is to increase the volume of air to be handled and to decrease the temperature; thereby inevitably reducing the draft in the case of a chimney, but in the case of a fan of proper size fortunately tending to increase the suction, unless the volume be in excess of the capacity of the fan. This results because, with a fan at constant speed, the intensity of the draft increases as the temperature of the gases passing through it is reduced. Coincident with the reduction of temperature is an increase in the weight of the gases — and hence of the admitted air — handled by the fan without change in speed. With forced draft beneath the grates the opposite tendency is noticeable, and a more or less direct loss of heat is the result. Under any condition the leakage naturally increases with the draft pressure, no matter how produced, and under equal pressures is obviously no greater with mechanical than with chimney draft.

An indication of the frequent amount of such leakage is shown in the results of the chemical analyses of samples of gas taken respectively from the back end of a boiler just before entering the tubes and from the uptake flue, as presented in Table No. 106. These results are sufficient to show the important necessity of preventing such loss.

Table No. 106. — Air Leakage through Boiler Settings.

CONDITIONS.	Coal consumed per Square Foot of Grate per Hour. Pounds.	Percentage of Air which leaked in between Back End and Uptake Flue.
Middle boiler in operation, dampers on other two boilers closed and packed,	21.45	22.08
Conditions the same	16.45	27.59
Three boilers in operation	15.09	15.40

The plant consisted of three boilers in one battery. In the first two tests an inward movement of air was perceptible at the doors of the outside boilers not in use, although the uptake dampers had been very carefully packed. The decreased leakage in the third case is principally due to the reduction of the ratio between exposed surface of the setting and the volume of air passing through the uptake flue. The original introduction of sheet-metal stops in the setting of such boilers is a comparatively simple matter, and if carefully carried out practically prevents all leakage. The lack of such arrangements, however, tends, in many instances, to render misleading the results of tests in which allowance has not been made for the effects of the incidental leakage.

CHAPTER IX.

CHIMNEY DRAFT.

Principles of Chimney Draft. — If two chimneys of identical construction and dimensions be connected at the bottom by a passage of the same cross-sectional area, and one of them be provided at its base with a means of heating the air, a definite air movement will result as soon as heat is applied. The cold air in one chimney, being heavier than the heated air in the other, will constantly seek to secure equilibrium of weights and pressures by flowing downward to the base of the heated chimney. As a natural consequence, the heated air will be forced upward and the cold air which takes its place will, in turn, be heated and follow the same course. A continuous flow will thus be maintained, its velocity and consequent volume being dependent upon the difference in density of the two columns of air; that is, upon the pressure difference. Although the difference in density results from the application of heat, the air movement is purely mechanical in its character, and depends directly upon the action of gravity.

The total difference in pressure upon the internal bases of the two chimneys is exactly equal to the difference in weight of the two columns of air within them. The relative difference may be expressed in any convenient terms, as pounds per square foot, ounces per square inch, or by the height of a column of water, mercury or other fluid necessary to balance this pressure. If a simple U-tube pressure gauge be partly filled with water, one end connected to the base of the cold chimney, and the other to the base of the hot chimney, the preponderance of weight of the air in the former will force the water downward in that arm of the tube and cause a corresponding rise in level in the other arm. The total difference in level may be read in inches of water and then readily resolved into ounces per square inch, or pounds per square foot.

Evidently, there being no difference between the character of the air in the cold chimney and that in the surrounding atmosphere, the same relative pressures will exist, and the same flow will continue if the cold chimney be removed and the air be allowed to directly enter the base of the hot chimney. Furthermore, the relative differences in the density and pressure created, being measured respectively by unit volume and unit area, are independent of the cross-sectional area of the hot chimney. In other words, the pressure difference is dependent only upon the height of the chimney and the difference in density between the heated air within and the cold air without. The terms "hot" and "cold" are, of course, only relative, for the draft is primarily dependent upon the actual temperature difference.

That changes in the temperature, either of the external atmosphere or the gases within the chimney, have a most marked influence upon the draft is very clearly shown by Table No. 107, in which the draft, as indicated in inches of water, is given for a chimney 100 feet high, with various internal and external temperatures. For any other height of chimney than 100 feet the height of the water column is directly proportional to that of the chimney. Hence doubling the height doubles the draft. This is not to be confused with the fact that the velocity which the draft has the power to create and the corresponding volume of air moved vary as the square root of the height. This table clearly indicates the necessity of high chimney temperatures for ample draft, and readily accounts for the stronger draft which exists in cold weather because of the greater temperature difference.

Table No. 107.—Height of Water Column Due to Unbalanced Pressures in Chimney 100 Feet High.

Temperature in Chimney.	Temperature of External Air.										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200°	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534

For a full comprehension and application of the principles of chimney draft it is necessary to consider them mathematically. If h be the height of the chimney, d the density of the external air, and d_1 that of the heated air, the pressure difference p for unit area may be expressed as —

$$p = hd - hd_1 = h\,(d - d_1)$$

The height of a column of external air which would produce this pressure, acting simply by its weight, may be found by dividing the pressure by the density of the external air. Therefore, if H represents the height of such a column, the expression will be —

$$H = \frac{p}{d} = \frac{d - d_1}{d}$$

The theoretical velocity with which the external air would enter the chimney, if no resistance existed, may be expressed by the equation —

$$v = \sqrt{2gH}$$

$$= \sqrt{2gh\left(\frac{d - d_1}{d}\right)}$$

In which v = velocity, in feet per second.
g = the acceleration due to gravity = 32.16.
H = head, or distance fallen, in feet.

The preceding deductions are based upon the assumption that there is no resistance to the movement of the air. Such a condition evidently cannot exist in the operation of the ordinary chimney. The motion of the gases creates a certain back pressure due to their friction on the inner surface of the chimney. This back pressure must be deducted from that due to the difference in density in order to ascertain the effective pressure which may be applied to compel the air to pass through a boiler furnace and up the chimney.

The effective pressure which in any given case is necessary to overcome the frictional and other resistances of the air in its passage through the fuel, the boiler tubes and other portions of the boiler, and impart to the gases the necessary velocity of entrance and exit, must depend upon the existing conditions of character of fuel, design and proportions of boiler, and other considerations which render exact determination extremely difficult.

Chimney Design. — From the preceding it is obvious that a chimney must be so designed as to create sufficient draft or pressure difference to overcome all resistances, and in addition impart the necessary velocity to the required amount of air. The number of pounds of coal which can be burned in a given time on a given grate equals the weight of air forced through, divided by the number of pounds of air required for the combustion of a pound of coal under the given conditions. As already shown, this latter amount may vary all the way from that theoretically necessary up to an amount 100 per cent or more in excess thereof. The weight of the air forced through in the given period of time is equal to the area through which it is admitted, multiplied by its velocity and by its density or weight per unit of volume. To impart this velocity, and also that through the furnace, boiler tubes, smoke flue and chimney, there is necessary a pressure which may be approximately ascertained by calculations based upon theoretical considerations. The additional pressure required to overcome the various resistances does not, however, admit of the theoretical determination, and can only be found by direct experiment, as has already been pointed out. These resistances are proportional to the square of the actual velocity, and depend on the diameter and length of tubes, flues and chimney, the thickness of the fuel and its state of division. It is, therefore, obvious that the proper design of a chimney to meet given conditions must be based upon the results of experiments under similar conditions. This fact readily accounts for the great divergence in the formulæ which have been presented, and for the necessity of a series of constants practically determined for application under stated conditions.

The earlier formulæ deduced by Peclèt, Rankine, Morin and Weisbach have until recently been generally accepted, but the theory upon which they are based has of late been the object of considerable criticism. Peclèt represented the law of draft by the formula —

$$h = \frac{u^2}{2g}\left(1 + G \times \frac{fl}{m}\right)$$

In which h = "head" or height of hot gases which, if added to the column of gases in the chimney, would produce the same pressure at the furnace as a column of outside air, of the same area of base, and height equal to that of the chimney.

u = required velocity of gases in chimney.

G = a constant to represent resistance to the passage of air through the coal.

l = length of the flues and chimney.

m = mean hydraulic depth, or the area of cross-section divided by the perimeter.

f = a constant depending upon the nature of the surfaces over which the gases pass, whether smooth, or sooty and rough.

Rankine's and the other formulæ are somewhat similar in form. The impossibility of assigning proper values to the constants in these formulæ has, up to the present time, prevented their practical application for chimney design, and resort has, as a consequence, been made in most cases to empirical methods.

Based upon this theory, however, the usual formula for determining the number of pounds of coal which may be burned per hour, under the conditions of the draft which may be created by a given chimney, takes the simple and practical form of —

$$F = CA\sqrt{H}$$

In which $F =$ number of pounds burned per hour.
$A =$ area of cross-section of chimney.
$H =$ height of chimney.
$C =$ coefficient varying from 10 to 20, according to the conditions.

As it stands, the formula implies that it is a matter of indifference in regard to the draft whether, for instance, — other things remaining the same, — a chimney is built 81 feet high and 2 feet square inside, or 16 feet high and 3 feet square inside; for in either case the product $A\sqrt{H}$ is the same. Common sense at once proves the impracticability of such a formula thus broadly applied. It is, therefore, usually qualified by limiting its application to cases when the total grate area is about eight times that of the chimney. But evidently such limitation is arbitrary, for this ratio of grate to chimney area is not constant for all plants.

The generally accepted empirical formulæ of Mr. William Kent,[1] which have been very generally adopted in the design of chimneys, are based upon the observed conditions in a large number of boiler plants, and take the general primary form of —

$$A = \frac{0.06F}{\sqrt{h}}$$

$$h = \left(\frac{0.06F}{A}\right)^2$$

In which $A =$ area of chimney.
$h =$ height of chimney.
$F =$ pounds of coal burned per hour.

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[1] Transactions American Society of Mechanical Engineers, Vol. VI., and Mechanical Engineers' Handbook, William Kent, New York, 1895.

The basis data are —

1. The draft power of the chimney varies as the square root of the height.

2. The retarding of the ascending gases by friction may be considered as equivalent to a diminution of the area of the chimney, or to a lining of the chimney by a layer of gas which has no velocity. The thickness of this lining is assumed to be 2 inches for all chimneys, or the diminution of area equal to the perimeter multiplied by 2 inches (neglecting the overlapping of the corners of the lining). For simplifying calculation, the coefficient is taken as the same for square and round chimneys, making the effective area E, as expressed by the equation, —

$$E = A - 0.6\sqrt{A}$$

In which A = total area.

3. The power varies directly as this effective area E.

4. A chimney should be proportioned so as to be capable of giving sufficient draft to cause the boiler to develop much more than its rated power, in case of emergencies, or to cause the combustion of 5 pounds of fuel per rated horse-power of boiler per hour.

5. The power of the chimney varying directly as the effective area E, and as the square root of the height H, the formula for horse-power of boiler for a given size chimney will take the form, $\text{H. P.} = CE\sqrt{H}$, in which C is a constant. Adopting the general value for C of 3.33, as determined from the results of numerous examples in practice, the formula for horse-power becomes —

$$\text{H. P.} = 3.33\,(A - .6\sqrt{A})\sqrt{H}$$

from which the values of E and H may also be obtained. In proportioning chimneys by these formulæ the height is generally first assumed, with due consideration of the conditions, and then the area for the assumed height and horse-power is calculated. The results of calculation for all ordinary dimensions of chimney and ranges of power are presented in Table No. 108.

The capacity in horse-power of a given chimney is inversely proportional to the amount of coal required per horse-power. Therefore, in large plants, where the economy of coal consumption is high, the capacities given in the table will be proportionably increased.

Prof. H. B. Gale,[1] having satisfied himself, both by experiment and mathematical analysis of the incorrectness of the common theory of chimney draft upon

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[1] Theory and Design of Chimneys. Horace B. Gale. Transactions American Society of Mechanical Engineers, Vol. XI.

Table No. 108.—Capacity in Horse-Power of Chimneys for Steam Boilers.

Diameter in Inches.	Side of Equivalent Square in Inches.	Effective Area $E=A-0.6\sqrt{A}$ in Square Feet.	HEIGHT OF CHIMNEY IN FEET.													
			50	60	70	80	90	100	110	125	150	175	200	225	250	300
18	16	0.97	23	25	27	29										
21	19	1.47	35	38	41	44										
24	22	2.08	49	54	58	62	66									
27	24	2.78	65	72	78	83	88									
30	27	3.58	84	92	100	107	113	119								
33	30	4.48		115	125	133	141	149	156							
36	32	5.47		141	152	163	173	182	191	204						
39	35	6.57			183	196	208	219	229	245						
42	38	7.76			216	231	245	258	271	289	316					
48	43	10.44				311	330	348	365	389	426					
54	48	13.51					427	449	472	503	551	595				
60	54	16.98					536	565	593	632	692	748				
66	59	20.83						694	728	776	849	918	981			
72	64	25.08						835	876	934	1,023	1,105	1,181	1,253		
78	70	29.73							1,038	1,107	1,212	1,310	1,400	1,485	1,565	
84	75	34.76							1,214	1,294	1,418	1,531	1,637	1,736	1,830	2,005
90	80	40.19								1,496	1,639	1,770	1,893	2,008	2,116	2,318
96	86	46.01								1,712	1,876	2,027	2,167	2,298	2,423	2,654
102	91	52.23								1,944	2,130	2,300	2,459	2,609	2,750	3,012
108	96	58.83								2,090	2,399	2,592	2,771	2,939	3,098	3,393
114	101	65.83									2,685	2,900	3,100	3,288	3,466	3,797
120	107	73.22									2,986	3,226	3,448	3,657	3,855	4,223
132	117	89.18									3,637	3,929	4,200	4,455	4,696	5,144
144	128	106.72									4,352	4,701	5,026	5,331	5,618	6,155

For pounds of coal burned per hour for any given size of chimney, multiply the figures in the table by 5.

certain points, substitutes a theory in which the height is dependent upon the stack temperature and the rate of combustion, and whose practical application is based upon certain experimentally determined constants.

He divides the problem of chimney design into two parts: "first, that of ascertaining the draft pressure necessary to burn the desired quantity of fuel in the furnace; second, that of determining the dimensions of a chimney which will produce the required draft at the least expense." The mean velocity, v, with which the air enters the various openings to the furnace he shows to be equivalent to —

$$v = \frac{T_a BF}{140{,}400\,a}$$

In which T_a = absolute temperature of outer air,
B = number of pounds of air supplied per pound of fuel,
F = number of pounds burned per hour,
a = total area of openings through which air is admitted to the fire,

when the relative density of the external air and the gases at the same temperature is in the proportion of 40 to 39. As the effective pressure varies as the square root of the corresponding velocity, the actual pressure difference, or frictional resistance P, is expressed by —

$$P = Kv^2$$

In which K = aggregate coefficient of resistance of fire grate, bed of coals, combustion chamber, boiler tubes, flue and chimney.

Prof. Gale has, by somewhat limited experiments, established the value of K as approximately 0.2 for average conditions, but this value varies greatly in different cases.

If in the expression $P = Kv^2$ the value of v (already given) be substituted, the equation becomes —

$$P = K\left(\frac{T_a BF}{140{,}400a}\right)^2$$

which indicates the reduction of pressure in pounds per square foot required at the bottom of the chimney which is to burn F pounds of fuel per hour, in a furnace having an area for air admission a (in square feet), and a coefficient of resistance K, allowing B pounds of air per pound of fuel.

The chimney height, H, to produce this draft pressure is shown to be —

$$H = \frac{KT_s}{T_s - 533 - \frac{M}{A^3}\left(\frac{T_s F}{15{,}000}\right)^2} \; \frac{F}{(3.5a)}$$

In which T_s = absolute temperature of chimney gases.
M = inside perimeter of chimney, in feet.
A = area of cross-section of chimney, in square feet;

and the other designations remain as in the preceding formulæ.

In this formula allowance is made for difference in density of external air and chimney gases, and in the form of cross-section. The temperature of the external air is assumed at 59°, and the most economical proportions of a chimney taken to be such that its cost is proportional to the $\frac{1}{3}$ power of the area multiplied by the $\frac{3}{2}$ power of the height.

From the last formula is ultimately derived the expression —

$$A = 0.07F^{\frac{3}{4}}$$

as an approximate formula for determining the most economical area for a chimney which is to burn F pounds of coal per hour. For chimneys of ordinary proportions the sectional area in square feet should by this formula be approximately equal to the number of pounds of fuel to be burned per minute.

For those cases in which the temperature of the chimney gases is between 150° and 600° the formula for height may be reduced to —

$$H = 100\,\frac{K}{t}\left(\frac{F}{a}\right)^2$$

In which t = absolute temperature of chimney gases.

This formula may be still further simplified by adopting a value for a equal to one-third of the grate area, making $K = 0.2$, and letting G represent the area of the grate in square feet, whereby —

$$H = \frac{180}{t}\left(\frac{F}{G}\right)^2$$

In this formula the height of chimney is proportional to the square of the rate of combustion per square foot of grate divided by the common temperature of the chimney gas. It is to be considered, however, as only a rough approximation. The value of K above must of necessity be largely affected by the character of the fuel, and further experiment is necessary to determine its value under given conditions.

Prof. C. A. Smith's formulæ are —

$$A = \frac{0.0825F}{\sqrt{h}}$$

$$h = \left(\frac{0.0825F}{A}\right)^2$$

in which the letters have the same designation as in the preceding formulæ. These are of the same form as Kent's, but with a different value for the constant. As a consequence, Smith's formula for area gives, for a stated height, larger results than Kent's for small fuel consumption; while for a large consumption the area by Smith's formula is greater than that by Kent's. The latter possesses the advantage of recognizing the practical fact that for larger powers the area of chimney required per horse-power becomes less.

The difficulties in the way of a purely theoretical consideration of the subject of chimney draft have been pointedly considered by Mr. William Kent,[1] who concludes with these words: "In the present state of our knowledge I do not think any satisfactory theory of chimneys can be framed which will include a consideration of all the different variables that affect the rate of combustion, and from which a formula can be derived that will prove of value in practice. . . . The best chimney formula that can be obtained at present is an empirical one, which may be modified or divided into two or more to meet different practical conditions, as new data are obtained from experiment."

The preceding discussion is sufficient to show the difference in formulæ and results, but is evidence that they would be in substantial accord if the constants could be based upon sufficiently exhaustive tests on the amount of grate, fuel and boiler resistances under all conditions of practice and qualities, kinds and conditions of fuel.

Evidently, the latter factor, that of the fuel, as has already been shown in the discussion of draft, is the most important of all, and any formula which does not make proper allowance for variation in it is liable to give uncertain results. Kent's constant is based upon an assumption of fair practice, which reduces to 15 pounds of coal consumed per square foot of grate per hour, under the draft created by a chimney 80 feet high and 42 inches in diameter. But the character of the coal is not specified.

Prof. W. P. Trowbridge,[2] basing his calculation on somewhat different data, determined the average rates of combustion for various heights of chimney, as here presented in Table No. 109. The air supply per pound of fuel was assumed at 250 cubic feet for all rates, and no attempt was made to indicate the kind of fuel or the effect of any change in its character. It will be noticed that this table gives a rate of 16.9 pounds for an 80-foot chimney, as compared with Kent's figure of 15 pounds; while Table No. 105, presented in the preceding chapter, makes the rate 19 pounds.

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[1] Transactions American Society of Mechanical Engineers, Vol. XI., p. 995.

[2] Heat and Heat Engines. W. P. Trowbridge. New York, 1874.

Table No. 109. — Height of Chimney to Produce Certain Rate of Combustion.

Height of Chimney in Feet.	Pounds of Coal Burned per Hour per Square Foot of Cross-Section of Chimney.	Pounds of Coal Burned per Hour per Square Foot of Grate, the Ratio of Grate to Cross-Section of Chimney being 8 to 1.
20	60	7.5
25	68	8.5
30	76	9.5
35	84	10.5
40	93	11.6
45	99	12.4
50	105	13.1
55	111	13.8
60	116	14.5
65	121	15.1
70	126	15.8
75	131	16.4
80	135	16.9
85	139	17.4
90	144	18.0
95	148	18.5
100	152	19.0
105	156	19.5
110	160	20.0

Prof. R. H. Thurston[1] gives this rough rule for the case of anthracite coal: "Subtract 1 from twice the square root of the height, and the result is the rate of combustion." This also gives the rate for an 80-foot chimney as 16.9.

As changes in the character of the fuel are likely to be made at any time in the life of a boiler plant, it is obvious that the chimney height, which is a measure of the draft pressure, should be made sufficient to meet all requirements. How much effect the kind of fuel may have upon the height of chimney necessary for its combustion is evidenced in the results of extensive tests with telescopic stacks by J. J. Kinder. He found that, for substantially equivalent results, the height for free-burning bituminous coals should be 75 feet, for slow-burning bituminous 115 feet, and for fine anthracite coals 125 to 150 feet.

These results point most clearly to the necessity, as already stated, of first determining the height of the chimney necessary to burn a given kind of fuel at a stated rate per square foot of grate, and then making it of sufficient area to burn the requisite amount. Evidently, in the light of common experience, as plainly shown by Mr. Kinder's tests, the first cost and the fixed charge for the necessary chimney will vary with the kind of coal used. The existing relation may be approximately illustrated by a consideration of Table No. 108. It is there indicated that 245 horse-power of boilers may be served either by a chimney 90 feet high and 42 inches in diameter, or by one 125 feet high and 39 inches in diameter. Such differences are well within the limits for different kinds of coal, as given above. If, then, in one case the coal can be burned with the draft produced by a 90-foot chimney, while in another a different kind of coal requires a chimney 125 feet high, the difference in the cost of the two chimneys must enter as an important item in the question of ultimate economy.

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[1] Manual of Steam Boilers. R. H. Thurston. 1888.

Upon the basis established by Prof. Gale, and previously referred to, the cost of a chimney is nearly proportional to the $\frac{1}{3}$ power of its area multiplied by the $\frac{3}{2}$ power of the height. As the area is proportional to the square of the diameter, the relative costs in the case of these chimneys become—

$$\text{Cost of 90-foot chimney} = 42^{\frac{2}{3}} \times 90^{\frac{3}{2}} = 10{,}316.$$
$$\text{Cost of 125-foot chimney} = 39^{\frac{2}{3}} \times 125^{\frac{3}{2}} = 16{,}072.$$

That is, the chimney would cost about 56 per cent more in one case than in the other to secure the same result in boiler power.

As regards the capacity of a chimney of a given height, which is directly dependent upon its area, it is necessary to make it originally sufficient for all future probabilities. If this capacity be made too great for present conditions, it becomes necessary to reduce the volume by means of dampers. But by no means, in the case of a chimney, can the draft itself, as measured by pressure difference and power to overcome resistances, be increased above that normally due to its height with given temperature conditions.

Thus, for instance, if with a given chimney the grate area be reduced and the rate of combustion proportionally increased so as to maintain the same total consumption, the resistances will be increased because of thicker fires and decreased free area through them. To overcome these resistances there is demanded greater draft, but as the draft of a chimney is absolutely limited by its height, any further expenditure of its draft for this purpose must by just so much reduce the portion of its draft available for producing the requisite air flow. If the chimney be large for the boiler plant, and if under the lower combustion rate it be necessary to operate it with practically closed dampers, there may be sufficient reserve to meet the requirements. But under ordinary conditions the chimney has but little power to respond to such requirements, or even to a material increase of the rate of combustion upon the regular grates. That is, its ability and capacity are distinctly limited, and to provide against contingencies it is usually necessary to build and pay the fixed charges on a structure more expensive than is regularly required.

On the other hand, with mechanical draft produced by means of a fan, as ordinarily applied, any increase in the resistances automatically increases the pressure difference which is created by the fan up to the limit of its capacity, as will be explained at length in a subsequent chapter. In other word, whereas with a chimney increased resistances decrease the pressure difference, the exact reverse is the tendency with a fan. As it has already been shown that in ordinary boiler practice almost all of such difference in pressure is required to overcome these resistances, the special adaptability of the fan for draft production cannot fail to be obvious.

Efficiency of Chimneys. — The chimney as a means of creating a movement of air depends upon the heating of that air; although, as shown, its actual movement is to be considered upon purely mechanical grounds. The heat thus employed is, however, absolutely wasted, so far as its utilization for any other purpose is concerned. Any attempt to extract more of the heat from the gases as they escape from the boiler must result in a reduction of the draft. This inherent loss is, therefore, always chargeable to any plant in which the draft is produced by a chimney, and possibilities in the way of increased economy must relate only to other losses so long as a given chimney is retained.

The percentage of the total calorific value of coal which is carried off by the products of combustion, and therefore available only for the production of draft, has already been presented in Table No. 52 for different degrees of excess of air and of temperature above the atmosphere. As there shown, this loss actually amounts to 19 per cent when the gases are at 500° and the excess of air is 100 per cent. Evidently such a great loss as is thus possible should require energetic effort to secure its reduction by a more economical substitute for the chimney.

Of course the weight of air moved by means of a chimney of given height must depend upon its area. As heat is the means by which this air movement is brought about, the efficiency of the chimney must be measured by the amount of heat expended for this purpose. Heat being transformable into work, the efficiency is, therefore, to be measured by the number of foot-pounds of work represented by the pressure difference exerted through the distance represented by the height of the column of cold air necessary to produce the given pressure, as compared with the number of foot-pounds represented by the total amount of heat expended.

Suppose, for the purposes of illustration, a chimney 100 feet high, having a cross-sectional area of 10 square feet, the atmospheric temperature at 62° and the temperature of the chimney gases at 500°; and further, for simplicity, assume that no work is lost in friction and that heated air is substituted for the hot gases, for their density and specific heat are approximately the same. Under these conditions the density d at 62° will be 0.0761 pounds per cubic foot, and the density d^1, 0.0414 at 500°. Therefore, by the formula previously given the pressure difference with the chimney 100 feet high will be —

$$\begin{aligned} p &= h\,(d - d^1). \\ &= 100\,(0.0761 - 0.0414). \\ &= 3.47 \text{ pounds per square foot.} \end{aligned}$$

This makes the total pressure difference $3.47 \times 10 = 34.7$ pounds over the entire area of the chimney.

The height of a column of external air which will produce the above pressure per square foot is —

$$\begin{aligned} H &= h\left(\frac{d - d^1}{d}\right) \\ &= 100\left(\frac{0.0761 - 0.0414}{0.0761}\right) \\ &= 45.6 \text{ feet.} \end{aligned}$$

The velocity of the air entering the base of the chimney under this head is —

$$\begin{aligned} v = \sqrt{2gH} &= \sqrt{64.32 \times 45.6} \\ &= 54.2 \text{ feet per second;} \end{aligned}$$

and its weight per second, —

$$\text{Weight} = 54.2 \times 10 \times 0.0761 = 41.25 \text{ pounds.}$$

The movement of this air is the result of heating it from 62° to 500°; that is, through $500 - 62 = 438°$. As the specific heat of air under constant pressure is 0.2375, the total heat expended per second in moving 41.25 pounds is —

$$\begin{aligned} \text{Heat expended} &= 41.25 \times 438 \times 0.2375 \\ &= 4{,}291.0 \text{ B. T. U.} \end{aligned}$$

As one heat unit is equivalent to 778 foot-pounds of work, the work equivalent to the total amount of heat expended, and which goes to waste without performing useful work in heating the water, is —

$$\begin{aligned} \text{Work equivalent of heat} &= 4{,}291.0 \times 778 \\ &= 3{,}338{,}398 \text{ foot-pounds.} \end{aligned}$$

But the work actually done is the result of overcoming a total pressure of 34.7 pounds through a distance of 54.2 feet; that is,—

$$\text{Work actually done} = 34.7 \times 54.2 = 1880.7 \text{ foot-pounds.}$$

Therefore, the efficiency of the chimney is —

$$\text{Efficiency} = \frac{1880.7}{3,338,398} = 0.000563$$

That is, less than six ten-thousandths of the heat expended is represented by the work done. In practice the resistance of the chimney, the cooling of the gases in their passage up it and other causes combine to decrease even this extremely low efficiency.

If in the place of the chimney there be substituted a fan of proper size, arranged to be driven by a direct-connected engine, the efficiency with which it would move the above-stated volume of air under the given conditions, but without the chimney, may be calculated with reasonable accuracy. Evidently the efficiency of a fan thus applied is the resultant of the efficiencies of the steam boiler, the engine and the fan, together with the loss by friction in the apparatus. If the combined efficiency of the boiler and engine be taken as one-tenth, the efficiency of the fan at the low value of only five-tenths and the loss from friction as two-tenths, or the efficiency as regards friction eight-tenths, the resulting efficiency of the system will be —

$$\text{Efficiency} = 0.1 \times 0.5 \times 0.8 = 0.04.$$

That is, of the work done, or its equivalent in heat units expended to produce a given result, one twenty-fifth is actually applied for that purpose; the remainder is lost in the processes of transformation and transmission and in friction. This efficiency, which allows for loss by friction, as was not the case with the chimney, is —

$$\frac{0.04}{0.000563} = 71.05$$

times greater than that of the chimney.

It may be shown that the relative efficiency of a fan and a chimney is dependent upon the height of the chimney and not upon the difference in temperature, and that it varies inversely as the height of the chimney. Thus in the case of a chimney 75 feet high the fan would, upon the same basis as above, have an actual efficiency of —

$$\frac{100 \times 71.05}{75} = 94.7$$

times greater than the theoretical efficiency of the chimney, while at 200 feet high the fan would still have 35.5 times higher efficiency; but this improvement in favor of the chimney would be largely offset by its proportionally greater cost as compared with a fan.

All other questions aside, the fan is, therefore, above question far more economical than the chimney. This economy means that the surplus heat can be utilized and the gases reduced to a minimum temperature before they enter the fan.

Says Mr. W. H. Bryan,[1] in discussing the advantages of mechanical draft: "As a general rule, however, it may be safely stated that in this year of our Lord, 1896, the building of tall chimneys to secure draft simply advertises the owner's lack of familiarity with modern improvements or his want of confidence in results easily demonstrated."

The following expression of opinion by the well-known engineer, Mr. William O. Webber,[2] is most emphatic: "I wish to go on record as advocating mechanical draft, as having the following advantages: Less first cost, and the larger the plant the greater the proportion of this saving of first cost; secondly, the elasticity of this system, as any given size of fan and stack can be used for two or three times its original capacity, simply by the speeding-up of the draft fan; and, third, the fact that this induced draft is absolutely controllable and is regulated by the pressure of steam in the boilers and the demand upon the boilers for steam; and, finally, it is not influenced by atmospheric conditions. I think that in this latter connection few people realize, unless they have spent considerable time in a boiler room, how the barometric and wind pressures affect the draft of ordinary brick chimneys, and that it is more strongly noticeable where an independent small metallic stack is used for each boiler of sufficient height to make a draft in the ordinary manner.

"The advantages in favor of the mechanical draft are, that no matter what the weather conditions are, as good a draft can be obtained at one time as another.

"This system has been in use long enough now to have demonstrated its practicability and economy, and in my opinion the old, tall, expensive brick chimney is out of date."

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[1] Boiler Efficiency, Capacity and Smokelessness, with Low-Grade Fuels. Wm. H. Bryan. A paper read before the Engineers' Club of St. Louis, Oct. 21, 1896.

[2] Large Brick Chimneys versus Mechanical Draft and Small Stacks. William O. Webber. Power, New York, July, 1897.

CHAPTER X.

MECHANICAL DRAFT.

Definition. — The chimney has long stood as practically the only available means of producing draft, which, thus produced, has been commonly called "natural draft." If the chimney satisfactorily met all of the requirements for modern boiler practice, one would scarce expect to see a substitute proposed. But the very substitution of another means is the strongest evidence of the acknowledged inadequacy of the chimney, not to produce draft, but to do it as satisfactorily and economically as the substitute.

Primarily introduced for the purpose of increasing the rate of combustion, artificial draft was designated as "forced draft," its field of application being considered to begin where that of the chimney ended. By later refinements it has, however, become not only a means of assisting chimney draft, and of producing the conditions requisite to accelerated combustion, but it is now accepted as a convenient and efficient substitute for the chimney under all ordinary conditions. The extent of its success and adoption must evidently be a measure of such convenience and efficiency.

Artificial draft may be, and has been, produced by means of steam jets inducing a flow of air, by blowing engines, by air compressors, by positive rotary blowers and by fan blowers or exhausters. Although the practical success of the locomotive is largely due to Stephenson's introduction of the steam nozzle for draft production, it does not follow that the same method is applicable where the exhaust steam would not otherwise be wasted. The blowing engine, the air compressor and the rotary positive blast blower all possess disadvantages which render undersirable their adoption for this purpose. In fact, they have been introduced to only a very limited extent. The centrifugal fan has, however, been most extensively applied under all conceivable conditions, until it has become the symbol of artificial, or, as it may properly be designated, of mechanical draft, and is today the accepted substitute for the chimney.

Steam Jets. — The steam jet as a means of inducing a flow of air is usually constructed upon the injection principle. It has been applied in the chimney for inducing the air movement through the fuel, as well as in the ashpit for forcing therein a volume of air which is caused to pass upward through the fuel. In connection with the latter arrangement, steam jets have often been introduced to also deliver air above the fuel, frequently in the form of a number of finely divided streams designed to mix intimately with the gases arising from the fuel bed.

The introduction of steam in conjunction with the air, which results from the use of the steam jet, is often asserted to assist in keeping a fire free and open, particularly in the case of fine anthracite fuels. But, in so far as steam for this purpose may be necessary, it can be as well introduced in connection with a fan. Consequently, the merits of the steam jet, as compared with a fan, must rest solely upon the relative efficiency with which a given amount of air is supplied; or, as more simply measured, by the proportion which the steam required to operate the steam jet or fan bears to the total steam produced by the boiler in connection with which it operates. In either case the percentage of steam thus used is largely dependent upon the size of the plant, being greatest with the smallest plant.

Careful experiments, conducted at the New York Navy Yard,[1] to determine the best form of steam jet for producing forced draft in launch boilers, served to show the inefficiency of such devices for this purpose. The results of five series of tests are presented in Table No. 110. A different form of jet, indicated by a designating letter, was used in each case; its supply of steam being taken from a boiler separate from that to which the jet was applied. The percentages of steam are such — the maximum being 21.2 per cent — as to make the adoption of a steam jet out of the question when any other means of draft production can be employed. In jet C, which had a hole only one-sixteenth inch in diameter, and which was the most economical of all, the steam used was one pound in two minutes. The amount of steam required by a fan blower, as will be shown later, is under ordinary conditions from a fraction of

Table No. 110. — Results of Experiments upon Steam Jets at New York Navy Yard.

	Pounds of Water Evaporated per Hour.				
	A	B	C	D	E
In boiler making steam	463.8	580.0	361.25	528.5	545.00
In boiler supplying steam jet	97.5	120.0	30.00	63.2	76.25
Per cent of steam made as used in steam jet .	21.20	20.70	8.30	12.00	14.00

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[1] Annual Report of the Chief of the Bureau of Steam Engineering, U. S. Navy. 1890.

1 per cent up to a possible maximum of 3 or 4 per cent in small boiler plants or with uneconomical apparatus; and practically the whole of this expenditure of power, in the form of exhaust steam, may be subsequently utilized for heating or similar purposes.

The case of the steam jet may be briefly summarized thus: It has the advantage of costing very little to put in and keep in repair. Its disadvantages are: first, it requires a very large amount of steam to run it; second, it introduces a large amount of water or steam, all of which has to be heated and carried up chimney; third, unless very carefully managed there is a large development of carbonic oxide, hydrogen and marsh gas, due to dissociation of the water, which has a tendency to carry off a great deal of heat in the stack; fourth, the intensity of draft producible by this means is distinctly limited; and, fifth, and by no means least, the noise incident to its use is at times almost intolerable.

The comparative effect of the steam jet and the fan blower upon the composition of the chimney gases is well shown by a test by Mr. Eckley B. Coxe,[1] upon two adjoining sets of boilers using the same fuel and fired by the same men, as presented in Table No. 111. A was fitted with a steam jet and B with a fan blower and Coxe stoker. The losses indicated by the excessive presence of carbonic oxide (CO), hydrogen (H) and marsh gas (CH_4) in the case of the steam jet are such as to most emphatically indicate its inefficiency.

Table No. 111. — Comparative Gas Analyses with Steam Jet and Fan Blower.

Designation.	Conditions.	Constituents.				
		O	CO	CO_2	H	CH_4
A	With steam jet	0.30	13.15	8.20	11.08	2.00
B	With fan blower	1.70	0.40	16.80		

The relative merits of the fan and the jet are thus expressed by Mr. Coxe: "The fan is more expensive to install and may cost more to keep in order, but where the arrangements can be made to utilize the heat in the stack gases it is more economical so far as heat units used are concerned. It has one great advantage,— it is possible to at all times obtain the exact blast necessary to produce the best results in the furnace, which is very important."

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[1] Some Thoughts upon the Economical Production of Steam, etc. Eckley B. Coxe. Transactions New England Cotton Manufacturers' Association. 1895.

Fans.—The centrifugal fan, or fan blower, as an apparatus for producing draft, is no new thing. As applied for the purposes of ventilation it dates back to the sixteenth century, but as a substitute for or auxiliary to the chimney its first application appears to have been made early in the present century. In 1827, Edwin A. Stevens, of Bordentown, N. J., arranged a fan for forcing the air into the ashpits of the boilers on the steamer North America. In conjunction with his brother, R. L. Stevens, he subsequently experimented at considerable length with various methods of applying the fan. It is said that in 1824 John Ericsson fitted the British steamer Victory for forced draft by means of a fan, and it is certain that in 1830 the Corsair was so equipped by him.

But engine speeds and steam pressures were then low; the demand for accelerated combustion was not urgent, and experience had not been gained in the proper application of fans for forced draft. As a consequence this economic improvement, which was to mean so much in later days, was but very meagrely adopted.

The fact that these first applications were made on steam vessels indicates the natural adaptability of the fan for the purpose. It is, therefore, not surprising to find that the arrangement was again taken up, this time by the United States during and subsequent to the war of the Rebellion. Extensive tests were conducted by Chief Engineer B. F. Isherwood, but there remained still another stage in its progress toward general application, both on sea and land.

At about this time B. F. Sturtevant began to manufacture and introduce fans of various sizes for the acceleration of draft in stationary boilers, and many were installed throughout the country. These were almost universally applied for forcing the air into the ashpit, and at once found a ready market because of the advantages incident to their use in the burning of cheap grades of fuel, which had previously been impossible with the ordinary chimney draft.

The advent of the torpedo boat marked the further introduction of forced draft for marine boilers. In these small, compact vessels tall stacks were out of the question, but strong draft and the utmost steaming capacity per ton of weight were an absolute necessity. From success with these smaller boats it was but a natural step to those of larger size. In 1877 the French government equipped the advice boat La Bourdonnais for forced draft, and in 1882 a definite move was made in the British Navy by providing fans for the production of draft on the Satellite and Conqueror.

Shortly after, the United States Navy again took up this important element in the design of the modern naval vessel and introduced forced draft upon all of the vessels of the "new navy." Of the vessels thus equipped almost all are supplied with Sturtevant fans.

From the navy it was but a natural step to the merchant marine. The advantages of the fan for draft production were recognized, and it has been extensively introduced both in the lake and ocean steamships. The form of application has varied greatly. The especial desirability of fuel economy on shipboard results in many cases in the use of special forms of heat abstractors or air preheaters, whose very use is possible only where a fan is employed to produce the draft.

On land the process of development has, during the past few years, been simply phenomenal. From the under-grate forced-draft systems there has been a gradual change to over-grate induced systems, until the matter of mechanical draft engages the attention of every progressive engineer in the design of a steam plant. The various forms of application, together with the types of fans therefor, will be described at length in succeeding chapters.

Two types of fans exist. The first, known as the disc or propellor wheel, is constructed on the order of the screw propellor, and moves the air in lines parallel to its axis, the blades acting upon the principle of the inclined plane. The second, or fan blower proper, consists in its simplest form of a number of blades extending radially from the axis and presenting practically flat surfaces to the air as they revolve. By the action of the wheel the air is drawn in axially at the centre and delivered from the tips of the blades in a tangential direction. This type may be simply designated as the centrifugal fan, or, more properly, as the peripheral discharge fan.

The propeller, or disc fan, which is available for ventilating purposes when it acts against slight resistances, is practically useless as a means of draft production. The desired results can only be secured by the use of the peripheral discharge type, which for this purpose is usually enclosed in a case of such shape as to provide free movement for the air as it escapes at the periphery, and an outlet through which it is all delivered. The detailed construction of these fans, with illustrations of their various types, will be indicated in the succeeding chapter.

Although theoretically there should be a difference in the form of the wheels designed for creating pressure and creating vacuum, yet, in the common acceptation of the terms, the distinction between a blower and an exhauster is primarily one of adaptation rather than of construction; the normal use of a blower being to force air into a given space, while an exhauster is employed to remove air from an enclosure. For convenience of attachment of pipe connections, an exhauster is provided with an inlet on one side only; while a blower, being exempt from this requirement, is provided with two inlets, one upon either side.

Theory of Fans. — In operation, the peripheral discharge fan sets in motion the air within it, which, acting by centrifugal force, is delivered tangentially at the outer circumference of the wheel. Air rushes in at the axial inlet to fill the space between the blades, in which there is, by the centrifugal action, a tendency to form a vacuum. The degree of this vacuum is dependent upon the circumferential speed of the wheel; and the velocity of the air discharged, through an outlet of proper size, is substantially equal thereto.

It has already been shown that a certain ideal head is necessary to produce a given velocity. This head, which may be expressed in terms of the pressure and density, as $h = \frac{p}{d}$, is usually designated as the "head due to the velocity."

The pressure, of which this head is one of the factors, is understood to be that existing in an enclosed space from which the air escapes at a velocity expressed by the formula, $v = \sqrt{2gh} = \sqrt{2g\frac{p}{d}}$. But the pressure which this stream of moving air may exert upon any external surface with which it comes in contact may be different from that which existed in the reservoir and produced the given velocity. This external pressure is due to the impact and reaction of the air, and for a given velocity depends in quantity upon the size and form of the surface and the angle of incidence. Theoretically, if the density is considered constant, this pressure, in the case of a stream striking a plane surface at right angles, will be double that which produced the velocity; or, more clearly stated, in its relation to water, the reaction of such a stream is equal to the weight of a column of water whose cross-section is that of the stream and whose height is double that ($2h$) due to the velocity. If the plane surface be of proper dimensions and surrounded by a raised border, to prevent the ready escape of the water, the theoretical pressure will be four times the head due to the velocity. Other shapes will give other values. With air, its lack of viscosity and the partial vacuum formed on the back side of the plate influence the actual results.

In the attempt to force air at a given velocity through a given pipe, it is the province of the fan wheel, if employed therefor, to create within the fan case a total pressure above the atmosphere which shall be sufficient to produce the velocity and also overcome the resistances of the case and the pipe. If, however, the pipe be removed and the fan be allowed to discharge the air through a short and properly shaped outlet, the pressure necessary will, with an efficient fan, be substantially that required to produce the velocity. The method of determining the velocity due to any given pressure has already been explained and the results of calculation embodied in Tables Nos. 91 and 92. From the same formulæ, properly transposed, the pressure due to any given velocity or necessary to its creation may be determined. The pressure thus determined is properly that which it is the purpose of a fan, employed as a device for moving air, to create. What reactionary pressure this velocity may produce as the air escapes from the fan is, therefore, in this connection, a matter of secondary importance.

The velocity of the fan tips or circumference of the fan wheel which is necessary to produce a given velocity of flow through a properly shaped outlet within the capacity of the fan is substantially equal to that velocity of flow. If, therefore, the peripheral velocity of a given fan is that indicated by any given quantity in column 3 of Table No. 91, the resulting pressure for the production of velocity through an outlet of proper size and shape will be practically that which corresponds to the given velocity in the table.

From the basis formula employed in the calculation of this table, as well as from preceding discussions, it is evident that the pressure created by a given fan varies as the square of its speed. The volume of air delivered is, however, practically constant per revolution, and therefore is directly proportional to the speed. The volume discharged under given pressure and velocity through an opening of given effective area is presented in Table No. 91. From this the total volume of air discharged may be calculated for any other opening whose area is known.

The work done by a fan in moving air is represented by the distance through which the total pressure is exerted in a given time. As ordinarily expressed in foot-pounds, the work per second would, therefore, be the product of the velocity of the air in feet per second, the pressure in pounds per square foot, and the effective area in square feet over which the pressure is exerted. If W represents the work done, p the pressure, a the area and v the velocity, the expression for the work becomes —

$$W = pav$$

But it has previously been shown that —

$$v = \sqrt{2g\frac{p}{d}}$$

hence —

$$p = \frac{dv^2}{2g}$$

Therefore, the value of W becomes —

$$W = \frac{dav^3}{2g}.$$

from which it is evident that the work done varies as the cube of the velocity; that is, as the cube of the revolutions of the fan. The reason is evident in the fact that the pressure increases as the square of the velocity, while the velocity itself coincidently increases; hence the product of these two factors of the power required is indicated by the cube of the velocity.

As one horse-power is equivalent to 33,000 foot-pounds of work per minute, the horse-power for a given area of discharge in square inches, when the value of g is taken as 32.2, may be expressed by —

$$\text{H. P.} = \frac{60dav^3}{144 \quad 33{,}000 \times 64.4}$$

$$= \frac{dav^3}{5{,}100{,}480}$$

The horse-power required to move air under various pressures and velocities has already been calculated and presented in Table No. 91. As applied to the case of a fan, the quantities there given, being for one square inch of effective area, must be multiplied by the total effective area through which the fan discharges and also by a coefficient which is the reciprocal of the efficiency. This coefficient, which under favorable working conditions may be indicated by an efficiency as high as 75 per cent, must of course be dependent upon the circumstances under which the particular fan is to be operated. But under no conditions can any device move air with the proportionate expenditure of power indicated in Table No. 91; for, as already shown, this value does not include the losses due to frictional resistances of the machine itself and of the air in its movement through the machine and connecting outlet or pipe.

As the weight of the air is dependent upon its temperature and the barometric and hygrometric conditions, it is evident that the pressure exerted and the power required by a given fan may vary greatly even at constant speed. Under ordinary conditions, however, changes in the height of the barometer and the humidity of the atmosphere have no appreciable effect upon the pressure and power. But the density varies inversely as the absolute temperature, and, therefore, should enter as a factor even in calculations with reference to air at or about ordinary atmospheric temperatures, and must be taken into account when heated air or gases are handled. The importance of this statement has already been shown in the discussion of Table No. 93, and will be further considered in connection with Table No. 113.

By means of the curves in Fig. 13 are graphically illustrated the theoretical relations between the revolutions at which a fan is run and the volume which is discharged, the pressure which is created and the horse-power which is required.

These curves are based upon the facts that the volume varies directly as the speed, the pressure as the square and the horse-power as the cube of the speed. Thus it is shown by the curves that if the speed is doubled the volume is also

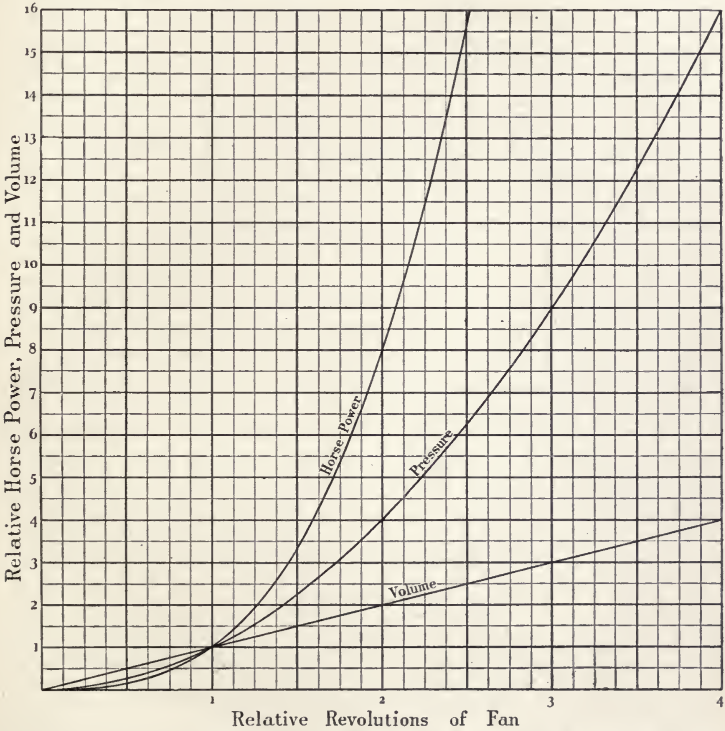

FIG. 13. THEORETICAL RELATIONS BETWEEN THE REVOLUTIONS OF A FAN, THE VOLUME DISCHARGED, THE PRESSURE CREATED AND THE HORSE-POWER REQUIRED.

doubled, the pressure is increased four times and the horse-power becomes eight times greater. The tremendous power expenditure which is required for even a moderate increase of speed of the fan is thus rendered very distinctly evident.

In selecting a fan, the facts presented in Fig. 13 should be borne in mind. It appears to be so simple to secure increased volume by running a given fan at higher speed that the influence upon the power required is frequently overlooked. If the necessary amount of power is actually furnished, its expenditure will entail great loss in efficiency as compared with that required to operate a fan properly proportioned to the work.

Design of Fans. — In the design of a wheel to meet given requirements it is necessary to make its peripheral speed such as to create the desired pressure, and then so proportion its width as to provide for the required air volume. Evidently, the velocity and corresponding pressure may be obtained either with a small wheel running at high speed or a large wheel running at slow speed. But if the diameter of the wheel be taken too small, it may be impossible to adopt a width, within reasonable limits, which will permit of the passage of the necessary amount of air under the desired pressure. Under this condition it will be necessary to run the fan at higher speed in order to obtain the desired volume. But this results in raising the pressure above that desired, and in unnecessarily increasing the power required. On the other hand, if the wheel be made of excessive diameter it will become almost impracticable on account of its narrowness. Between these two extremes a diameter must be intelligently adopted which will give the best proportions. If the fan is to be driven by a direct-connected engine, as is the common practice in mechanical draft, it may be further necessary to so adapt the number of revolutions, and consequently the peripheral velocity of the wheel, to a given engine, that its speed may not be excessive and that the proper amount of power may be generated.

The actual volumetric capacity of a given fan, operating under practical conditions, is naturally to be sought as a means of measuring it relatively to any other fan. But manifestly such capacity is somewhat difficult of pre-determination. In the case of a steam engine, its nominal rating — that by which one engine may be measured relatively to another — is based upon the diameter and stroke of the cylinder, the number of revolutions and the mean effective pressure. But the power thus calculated by no means represents the amount which may be delivered to a given machine, for the sole purpose of operating which the engine is employed. This latter amount will be less than that calculated, to the extent that power is absorbed in the internal friction of the engine and by the intermediate mechanism of transmission. So in the case of a fan wheel, its theoretical volumetric capacity will depend upon its dimensions and the speed at which it is operated. But in practice the actual amount of air delivered will also be largely dependent upon the fact of the wheel being encased, the character and dimensions of the case, and the size and resistances of the passages through which the air is conducted. The equivalent of such resistances is in boiler practice usually represented by the grates, the fuel, tubes, etc., and may evidently be so great at times as to very seriously reduce the theoretical air discharge of the fan.

Evidently, it is improper to compare fans when operating under such conditions that these resistances cannot be definitely determined. The simplest and most natural condition is that in which the fan is operated without other resistance than that of the case; that is, with open inlet and outlet. But for proper comparison of different fans, the areas through which the air is discharged should bear some constant relation to the dimensions of the wheels themselves.

It has been determined experimentally that a peripheral discharge fan, if enclosed in a case, has the ability, if driven to a certain speed, to maintain the pressure corresponding to its tip velocity over an effective area which is usually denominated the "square inches of blast." This area is the limit of its capacity to maintain the given pressure. If it be increased the pressure will be reduced, but if decreased the pressure will remain the same. As fan housings are usually constructed, this area is considerably less than that of either the regular inlet or outlet. It, therefore, becomes necessary, in comparing fans upon this basis, to provide either the inlet or the outlet with a special temporary orifice of the requisite area and proper shape, and make proper correction for the contracted vein. The fan is thus, in a sense, placed in a condition of restriction of discharge, which it approaches in practice only in so far as the resistances of pipes, passages and material through which the air must pass have the effect of reducing the free inlet or outlet of the fan.

The square inches of blast, or, as it may be termed, the capacity area of a cased fan, may be approximately expressed by the empirical formula, —

$$\text{Capacity area} = \frac{DW}{x}$$

In which $D =$ diameter of fan wheel, in inches.
$W =$ width of fan wheel at circumference, in inches.
x = a constant dependent upon the type of fan and casing.

The value of x has been very carefully determined by this Company for different types of fans, but these values must be applied with great discretion, acquired through experience and a thorough knowledge of all the conditions liable to affect the fan in operation. An approximate value of x for general practice is not far from 3, but this is to be used only to determine the capacity area over which the given pressure may be maintained. This is not a measure of the area of the casing outlet, which is always larger than the square inches of blast. As a consequence, the pressure is lower and the volume discharged is somewhat greater than would result through an outlet having the square inches of blast for its area. But the maximum pressure may be realized when the sum of resistances is equivalent to a reduction of effective outlet area to that of square inches of blast. The volume of air which under the given pressure will flow through the given capacity area, and hence the volumetric capacity of the fan under the given conditions, may be determined from Table No. 91. In a similar manner the horse-power may be ascertained, the proper efficiency coefficient being applied.

Both the volume and the power required will evidently increase with the area of the outlet, being greater with the normal outlet than with that representing the capacity area. But this increase will not be proportional to the area, for the pressure and consequently the velocity will be lower with the larger area. The greatest delivery of air and the largest consumption of power will occur when the casing is entirely removed and the fan left free to discharge entirely around its periphery.

Although the theoretical considerations which govern the design of fans have here been given, the conditions which exist in any given case must enter into any decision as to the practical dimensions to be given the fan.

If volume alone, regardless of pressure, is the requisite, the larger the fan, the less the power required. There is a strong temptation, however, for a purchaser to buy a smaller fan and run it at higher speed; for he sees only the first cost and does not realize the entailed expenditure for extra power. If possible, a fan should never be made so small that it is necessary to run it above the required pressure in order to deliver the necessary volume. To double the volume under such circumstances requires eight times the power; three times the volume demands twenty-seven times the power.

For certain purposes, such as the blowing of cupola furnaces, a comparatively small volume of air is required, but under high pressure. In steam-boiler practice the volume is relatively greater, and the pressure less; the general range of pressures required has already been shown. The former wheel requires to be narrow at the circumference, thus providing for the escape of only a small amount of air. When a fan is employed for exhausting hot air or gases, the speed required to maintain a given pressure difference is evidently greater than that necessary when cold air is handled, the difference being due to, and inversely proportional to, the absolute temperature.

From the last formula it is evident that if the diameter of the wheel be known, as determined by the previous considerations, its width may be determined for any given capacity area when the value of x is ascertained. If, furthermore, the pressure at which it is to operate be pre-determined, the number of revolutions may be readily ascertained by means of Table No. 91; for the tip speed must be equal to the velocity there given as corresponding to the stated pressure. From the same table the volume of air delivered and the theoretical horse-power required for the given capacity area may be determined. How much greater than these amounts the actual results will show must of course depend upon the circumstances, particularly the area of discharge existing in each case. For the more ready determination of the number of revolutions of any given size of fan which are necessary to produce a given pressure over the capacity area of the wheel, Table No. 112 has been prepared. This is self-explanatory. Being based on the values in Table No. 91, these results apply only to a temperature of 50° Fahr. For other temperatures due correction must be made.

In Table No. 93 has already been indicated the effect of changes in temperature; but, as there presented, they apply primarily to the effect upon the pressures required to produce given velocities. As it is the province of a fan to operate at certain velocities in order to produce desired pressures, the important temperature factors are presented in Table No. 113 in somewhat different form. The temperatures in the table are in degrees Fahrenheit above zero. These results are also graphically presented in Fig. 14, the curves representing the values in the columns as designated.

The basis upon which the various relations have been calculated is as follows, the atmospheric temperature being taken at 50° Fahr. and absolute zero at 461° below zero, Fahr.: —

Column 2. The volume of the same weight of air is directly proportional to its absolute temperature.

Column 3. The weight of the same volume of air is inversely proportional to its absolute temperature.

Column 4. The speed of the same fan necessary to handle the same weight of air is directly proportional to its absolute temperature.

Column 5. The pressure difference due to the speed of the same fan necessary to handle the same weight of air is directly proportional to the square of the speed and inversely proportional to the absolute temperature of the air.

Column 6. The pressure difference due to the same speed of fan handling the same volume of air is inversely proportional to its absolute temperature.

Column 7. The speed necessary to produce the same pressure difference is directly proportional to the square root of the absolute temperature.

Column 8. The power required for the same speed and volume is inversely proportional to the absolute temperature.

Table No. 112.—Revolutions of Fan of Given Diameter Necessary to Maintain a Given Pressure over an Area which is within the Capacity of the Fan.

Diameter of Fan Wheel, in Feet.	Pressure, in Ounces per Square Inch.												
	⅛	¼	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1¾
1	582	823	1,007	1,163	1,300	1,423	1,537	1,643	1,742	1,836	1,925	2,010	2,170
1¼	466	658	806	930	1,040	1,139	1,230	1,314	1,394	1,469	1,540	1,608	1,736
1½	388	549	672	775	867	949	1,025	1,095	1,162	1,224	1,284	1,340	1,447
1¾	333	470	576	665	743	813	878	938	996	1,049	1,100	1,149	1,240
2	291	411	504	582	650	712	769	822	871	918	963	1,005	1,085
2¼	259	366	448	517	578	633	683	730	774	816	856	893	964
2½	233	329	403	465	520	570	615	657	697	734	770	804	868
2¾	212	300	366	423	473	518	559	597	634	668	700	731	789
3	194	274	336	388	433	475	513	548	581	612	642	670	723
3½	166	235	288	332	372	407	439	469	498	525	550	574	620
4	146	206	252	291	325	356	384	411	436	459	481	502	543
4½	129	183	224	258	289	316	342	365	387	408	428	447	482
5	116	164	202	232	260	285	308	329	349	367	385	402	434
5½	106	149	183	211	236	259	280	299	317	334	350	366	395
6	97	137	168	194	217	238	256	274	290	306	321	335	362
6½	90	126	155	179	200	219	236	253	268	282	296	309	334
7	83	117	144	166	186	203	220	235	249	262	275	287	310
7½	78	110	135	155	173	190	204	219	232	245	257	268	289
8	73	103	126	146	163	178	192	205	218	230	241	251	271
8½	69	97	119	137	153	167	181	194	205	216	226	236	255
9	65	92	112	129	144	158	171	183	194	204	214	223	241
9½	61	87	106	123	137	149	162	173	183	193	203	212	228
10	58	82	101	116	130	142	154	164	174	184	193	201	217
11	53	75	92	106	118	129	140	150	158	167	175	183	197
12	49	69	84	97	108	119	128	137	145	153	160	168	181
13	45	63	78	90	100	110	116	126	130	141	148	155	167
14	42	59	72	83	93	102	110	117	124	131	138	144	155
15	39	55	67	78	87	95	102	110	116	122	128	134	145

Table No. 112.—Revolutions of Fan of Given Diameter Necessary to Maintain a Given Pressure over an Area which is within the Capacity of the Fan.—Concluded.

Diameter of Fan Wheel, in Feet.	Pressure, in Ounces per Square Inch.												
	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8
1	2,319	2,590	2,834	3,058	3,265	3,460	3,643	3,817	3,992	4,141	4,293	4,439	4,580
1¼	1,855	2,072	2,267	2,446	2,612	2,768	2,915	3,054	3,186	3,313	3,434	3,551	3,664
1½	1,546	1,727	1,889	2,039	2,178	2,307	2,429	2,545	2,655	2,761	2,862	2,960	3,053
1¾	1,325	1,480	1,619	1,747	1,866	1,977	2,082	2,171	2,276	2,366	2,453	2,536	2,617
2	1,159	1,295	1,417	1,529	1,633	1,730	1,822	1,909	1,996	2,070	2,146	2,219	2,289
2¼	1,030	1,151	1,259	1,359	1,451	1,538	1,619	1,696	1,770	1,840	1,908	1,973	2,035
2½	928	1,036	1,134	1,223	1,306	1,384	1,457	1,527	1,593	1,656	1,717	1,776	1,832
2¾	843	942	1,030	1,112	1,188	1,258	1,325	1,388	1,448	1,506	1,561	1,614	1,665
3	773	863	945	1,019	1,089	1,153	1,215	1,272	1,328	1,380	1,431	1,480	1,527
3½	662	740	810	874	933	989	1,041	1,086	1,138	1,183	1,226	1,268	1,308
4	580	647	708	764	816	865	911	954	998	1,035	1,073	1,110	1,145
4½	515	575	630	679	726	769	810	848	885	920	954	986	1,018
5	464	518	567	612	653	692	729	763	796	828	859	888	916
5½	422	471	515	556	594	629	662	694	724	753	781	807	833
6	386	432	472	510	545	577	607	636	664	690	716	740	763
6½	357	398	436	470	502	532	561	587	613	637	661	683	705
7	331	370	405	437	466	494	520	543	569	592	613	634	654
7½	309	345	378	408	435	461	486	509	531	552	572	592	611
8	290	324	354	382	408	432	455	477	499	518	537	555	572
8½	273	305	333	331	354	375	429	449	469	487	505	522	539
9	258	288	315	340	363	384	405	424	443	460	477	493	509
9½	244	273	298	322	344	364	384	402	419	436	452	467	482
10	232	259	283	306	327	346	364	382	398	414	429	444	458
11	211	235	258	278	247	315	331	347	362	376	390	404	416
12	193	216	236	255	272	288	304	318	332	345	358	370	382
13	178	199	218	235	251	266	280	294	306	319	330	341	352
14	165	185	202	218	233	247	260	271	284	296	307	317	327
15	155	173	189	204	218	231	243	254	266	276	286	291	305

Table No. 113. — Relation of Volume, Weight and Pressure of Air and Speed and Power of a Fan with Air at Different Temperatures.

Temperature, Degrees Fahr. 1	Volume for Same Weight. 2	Weight for Same Volume. 3	Speed of Fan to Handle Same Weight. 4	Pressure Difference due to Speed Necessary to Handle Same Weight. 5	Pressure Difference for Same Speed and Volume. 6	Speed to Produce Same Pressure Difference. 7	Power for Same Speed and Volume. 8	Power for Speed Necessary to Handle Same Weight. 9	Power Necessary to Handle Same Weight at Same Pressure with a Properly Proportioned Fan. 10
30°	0.96	1.04	0.96	0.96	1.04	0.98	1.04	0.92	0.96
40	0.98	1.02	0.98	0.98	1.02	0.99	1.02	0.96	0.98
50	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
60	1.02	0.98	1.02	1.02	0.98	1.01	0.98	1.04	1.02
70	1.04	0.96	1.04	1.04	0.96	1.02	0.96	1.08	1.04
80	1.06	0.94	1.06	1.06	0.94	1.03	0.94	1.12	1.06
90	1.08	0.93	1.08	1.08	0.93	1.04	0.93	1.17	1.08
100	1.10	0.91	1.10	1.10	0.91	1.05	0.91	1.21	1.10
125	1.15	0.87	1.15	1.15	0.87	1.07	0.87	1.32	1.15
150	1.20	0.84	1.20	1.20	0.84	1.09	0.84	1.43	1.20
175	1.24	0.81	1.24	1.24	0.81	1.11	0.81	1.55	1.24
200	1.29	0.78	1.29	1.29	0.78	1.14	0.78	1.67	1.29
225	1.34	0.75	1.34	1.34	0.75	1.16	0.75	1.80	1.34
250	1.39	0.72	1.39	1.39	0.72	1.18	0.72	1.93	1.39
275	1.44	0.69	1.44	1.44	0.69	1.20	0.69	2.07	1.44
300	1.49	0.67	1.49	1.49	0.67	1.22	0.67	2.22	1.49
325	1.54	0.65	1.54	1.54	0.65	1.24	0.65	2.36	1.54
350	1.59	0.63	1.59	1.59	0.63	1.26	0.63	2.51	1.59
375	1.63	0.61	1.63	1.63	0.61	1.28	0.61	2.68	1.63
400	1.68	0.59	1.68	1.68	0.59	1.30	0.59	2.84	1.68
425	1.73	0.58	1.73	1.73	0.58	1.32	0.58	3.01	1.73
450	1.78	0.56	1.78	1.78	0.56	1.34	0.56	3.18	1.78
475	1.83	0.55	1.83	1.83	0.55	1.35	0.55	3.35	1.83
500	1.88	0.53	1.88	1.88	0.53	1.37	0.53	3.56	1.88
525	1.93	0.52	1.93	1.93	0.52	1.39	0.52	3.71	1.93
550	1.98	0.51	1.98	1.98	0.51	1.41	0.51	3.92	1.98
575	2.03	0.49	2.03	2.03	0.49	1.43	0.49	4.12	2.03
600	2.08	0.48	2.08	2.08	0.48	1.44	0.48	4.32	2.08
625	2.13	0.47	2.13	2.13	0.47	1.46	0.47	4.54	2.13
650	2.18	0.46	2.18	2.18	0.46	1.48	0.46	4.75	2.18
675	2.22	0.45	2.22	2.22	0.45	1.49	0.45	4.93	2.22
700	2.27	0.44	2.27	2.27	0.44	1.51	0.44	5.16	2.27
725	2.32	0.43	2.32	2.32	0.43	1.52	0.43	5.39	2.32
750	2.37	0.42	2.37	2.37	0.42	1.54	0.42	5.62	2.37
775	2.42	0.41	2.42	2.42	0.41	1.56	0.41	5.86	2.42
800	2.47	0.40	2.47	2.47	0.40	1.57	0.40	6.10	2.47

Column 9. The power required to operate the same fan at the speed necessary to handle the same weight of air is directly proportional to the cube of the speed and inversely proportional to the absolute temperature.

Column 10. The power necessary to handle the same weight of air at the same pressure difference by means of a properly proportioned fan is directly proportional to the speed and to the area required for the passage of the given weight corresponding to the speed. That is, it is directly proportional to the absolute temperature.

The conditions indicated in column 10 are substantially those usually presented in induced-draft practice, wherein it is generally necessary to move the same weight of air under the same pressure, but at a higher temperature than would be required in the case of air handled for forced draft. From column 4 it is evident that if with the same fan it be attempted to handle the same weight at increased temperature, the speed must also be increased; and it is further shown, per column 9, that the power required under these circumstances rises very rapidly with the temperatures. But a fan so designed as to move the same weight without exceeding the pressure created at the lower pressure will require far less power.

Thus suppose that a fan is to be designed to handle air or gases (they both being here considered, for simplicity, as of the same specific gravity) at a temperature of 300°. At this temperature the speed necessary to maintain the same pressure is relatively 1.22, per column 7. But through a given area of opening, such as would represent the capacity area of a proper fan for the lower temperature, the actual weight moved would be the product of the velocity and the density; that is, only $1.22 \times 0.67 = 0.82$ at 300°. In order to maintain the weight at unity, the area of discharge for the given velocity must therefore be increased until it becomes $\frac{1.0}{0.82} = 1.22$. So that the power actually expended in moving the same weight with a properly designed fan would for the same pressure be, at 300°, the combined product of the unit pressure, the area and the velocity; that is, $1 \times 1.22 \times 1.22 = 1.49$ times that required for a fan properly designed to maintain the same pressure and handle the weight of air at 50°.

If the same diameter of fan be employed under each condition, and the greater volume, in the case of the higher temperature, be provided for by widening the fan, the power required per revolution will be in the relation of 1 at 50° to 1.22 at 300° temperature. For, as already shown, the total power required at the latter temperature is 1.49, while the speed is 1.22, both relatively to the conditions at 50°; hence the power per revolution is $\frac{1.49}{1.22} = 1.22$. From

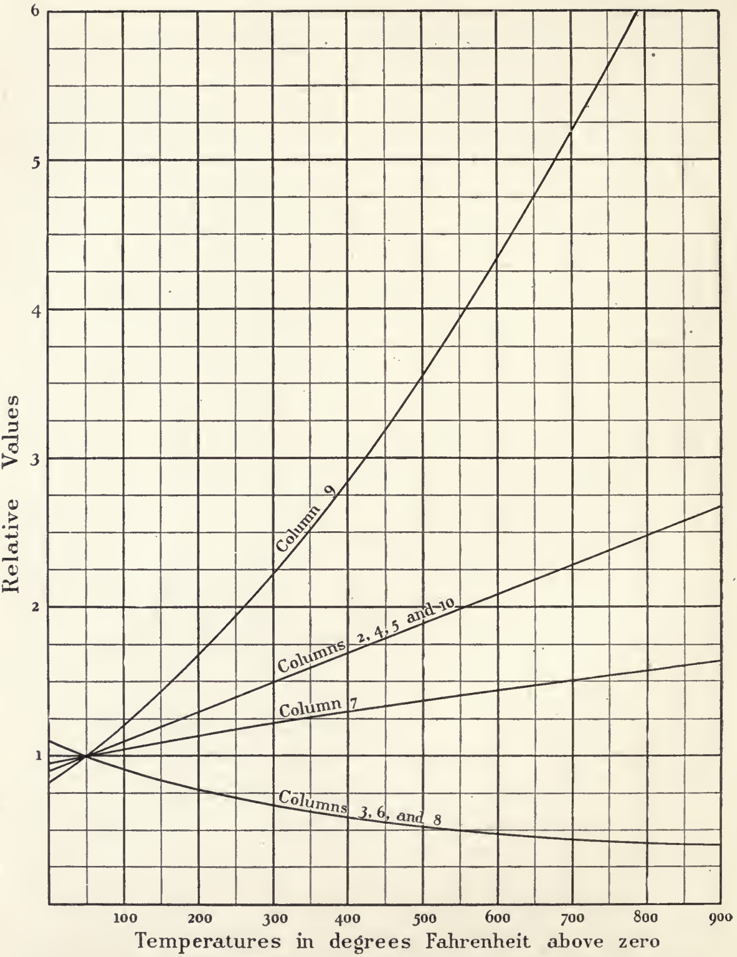

FIG. 14. RELATION OF VOLUME, WEIGHT AND PRESSURE OF AIR AND SPEED AND POWER OF A FAN WITH AIR AT DIFFERENT TEMPERATURES.

this it is evident that if in each case the fan be driven by a direct-connected engine, whose revolutions of necessity correspond to those of the fan, the engine will require additional piston area, or mean effective pressure, to the extent of only 22 per cent.

From Table No. 113 it is evident that, if a fan be designed to handle a given weight and corresponding volume of air at 50° under a given pressure and with the expenditure of a given amount of power, the following conditions will hold, if the temperature of the air be raised to 300° for instance. If the same weight is to be handled, the same fan will (per column 4) have to be run at 1.49 times the speed. At this speed the pressure difference produced will (per column 5) be 1.49 times, and the power expended (per column 9) 2.22 times, that under the first condition. For the same speed and volume the weight handled will (per column 3) be 0.67, the pressure (per column 6) 0.67 and the power (per column 8) 0.67 of that at a temperature of 50°; and to produce the same pressure difference the fan will have to run (per column 7) at 1.22 times the speed required at 50°. From this it is further evident that in the attempt to handle, with a given fan, the same weight of air at a higher temperature it is necessary to increase the speed above that required to produce the same pressure difference; and that the power expended is correspondingly and unnecessarily increased.

It is, therefore, obvious that the fan should be designed to meet the specific conditions. Thus a properly proportioned fan will produce the same pressure and handle the same weight of air at 300° with 1.49 times the power (per column 10) required to obtain the same results with the air at 50°; that is, with $\frac{1.49}{2.22} = 0.55 =$ only 70 per cent of that which would have been necessary had the same fan designed for 50° been used in both cases.

These are purely theoretical relations, but they hold substantially under practical conditions. The leakage of air through boiler settings and the decrease of efficiency through losses in power transmission, although they affect, yet do not properly enter into these relations, but must be provided for by additional capacity in fan and engine. Similarly, the increase of volume due to the products of combustion may be provided for.

Methods of Application. — The methods of application of mechanical draft may be broadly classified under two heads, — the plenum and the vacuum methods. Although both were experimented upon by Stevens in 1827 and in the succeeding years, yet the former remained for a long period practically the only form in which mechanical draft was applied. As the term implies, the air under the plenum method is forced through the fire; that is, the pressure maintained below the fire is greater than that of the atmosphere; hence the general term, "forced" draft.

Under the plenum method the air may be supplied in either of two ways. First, by making the ashpit practically air tight, and then forcing into it the air in sufficient quantity and under the requisite pressure. Evidently, the only escape for the air being through the fuel, it must all be utilized for the purposes of combustion. Second, by making the fire room itself practically air tight and maintaining therein the required air pressure by means of a fan of sufficient capacity to constantly make good the amount of air which under pressure passes to the ashpits and thence through the fuel.

Under the vacuum method there is practically only one means of application, — that by the introduction of an exhausting fan in the place of a chimney. This is commonly known as the "induced" or "suction" method. The fan thus serves to maintain the vacuum which would exist if a chimney were employed, and its capacity can be made such as to handle the gases which result from the processes of combustion. A short and comparatively light stack usually serves to carry these gases sufficiently high to permit of their harmless escape to the atmosphere.

Evidently, the method of application to be adopted must depend upon the circumstances. It cannot be said that under all conditions any one of these three principal methods, or their numerous modifications, is superior to the others. It is the primary purpose of this book to show the superior advantages of mechanical draft, and, secondarily, to point out, so far as possible, the conditions under which any given system may prove most efficient; but it is not the purpose to advocate any one system, whether patented or not, in preference to the others. This Company is interested in any system that embodies the use of a fan; for the successful application of a fan for such a purpose requires the extended experience which it has been the privilege of this Company to enjoy.

In the succeeding pages will be presented detailed descriptions of the various methods of application, and of the general advantages of mechanical draft.

Closed Ashpit System. — This system was naturally first applied because of its ready adaptability to existing conditions. And for the same reason it has, to a great extent, been the system introduced wherever, in an existing plant, it has been desired to burn a cheaper grade of fuel or to add to the steaming capacity. As most simply applied for stationary boiler work, the air has been introduced in the side of the ashpit setting through a pipe from the fan. There is a tendency, however, with such a simple arrangement, to fail to properly distribute the air in the ashpit. The result of unequal distribution, as may occur with improper introduction of the air, is a tendency to blow holes through the fire and overheat the grate bars wherever the draft is concentrated. Of course, the more intense the draft the greater this tendency. This may readily be overcome by properly deflecting the air by means of a Sturtevant Ashpit Damper, so as to insure its thorough distribution throughout the ashpit before it rises to the grate.

A convenient form of damper, which prevents the blowing of holes through the fire, is illustrated upon a succeeding page. It may be placed either in the bottom of the ashpit and arranged to receive its air from an underground duct in front of the boilers; or it may be located in the front of the bridge wall, with the hinge of the door or damper above, so as to deflect the air downward when it is opened. In either case the damper can be readily operated by a rod from the boiler front. This arrangement necessitates keeping the ashpit doors closed.

In marine boilers with internal fire boxes such arrangements are usually inadmissible. There is, however, a method of introduction through the back end of the ashpit, the air being conducted thereto from the back of the boiler by means of a passage specially provided and extending through the water back and combustion chamber. Ordinarily, however, the air is admitted through the ashpit doors or the openings provided for them. This necessitates a removable arrangement so that the ashpits may be cleaned. Numerous devices have been applied for this purpose, as is evident from the illustrations in a succeeding chapter.

The pressure within the ashpit and the furnace chamber causes all leakage to be outward. The tendency is, therefore, to blow the ashes out of the ashpit and the flame, smoke and fuel out of the fire doors, but with slight effect in the case of stationary boilers at moderate rates of combustion. In the marine service, in order to avoid inconvenience from this source, boilers are frequently fitted with false fronts, within which the air pressure is maintained. By a proper arrangement of double doors and dampers the disadvantage and danger from flame is completely overcome. The mere arrangement of dampers so connected to the doors that they close when the fire doors are opened is of great advantage. The false front further presents an excellent yet simple means of admitting air above the fuel, a feature which enters into most arrangements of this character. The conditions which have to be met in some cases of marine practice are exemplified in Table No. 114, in which are presented the results of three tests of the air pressures produced by Sturtevant fans on U. S. S. Swatara,[1] which was equipped on the closed ashpit system. Pressures are given in inches of water.

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[1] Annual Report of the Chief of Bureau of Steam Engineering, U. S. Navy. 1888.

Although the closed fire-room system has been far more extensively applied in the naval marine than has the closed ashpit system, it has been due largely to the existing conditions; and there can be no doubt that of the two the latter arrangement is more satisfactory when the conditions permit of its introduction. The controlling conditions are well expressed in these words of Engineer-in-Chief George W. Melville, U. S. N.:[1] "As between the two methods of forced draft in most common use, that by closed fire rooms and by closed ashpits, I am decidedly in favor of the latter when it can be applied. I make this proviso for the reason that some may at once ask why, if I am a believer in ashpit forced draft, nearly all of our large vessels, recently designed, have forced draft on the closed fire-room system. It is simply because, in a war vessel with a protective deck and minute water-tight sub-divisions, it is extremely difficult, where there are a number of large boilers, to so arrange the blowers for closed ashpit forced draft as to ventilate the fire room thoroughly. . . .. The San Francisco, of our navy, has ashpit forced draft, and all who have had experience on her and on other similar vessels speak in the highest terms of praise of the greater facility, convenience and comfort which attend this method.

Table No. 114.—Air Pressures in Connection with Boilers of U. S. S. Swatara.

CONDITIONS.	Number of Test.		
	1	2	3
Air pressure in conduit	2.87	4.76	3.48
Air pressure in ashpit	2.08	4.01	2.34
Air pressure in furnace-door frame	1.86	3.61	2.13
Air pressure in furnace	1.63	3.18	1.75
Air pressure in uptake	0.13	0.10	0.07
Revolutions of fan per minute	438.6	584.8	476.6

"It is to be noted, also, with this method of forced draft, that when there is any care at all taken in the fire room to keep the grate bars covered leaky tubes in the combustion chamber are unknown."

The absence of the protective deck, the possibility of open fire rooms and the greater space which is usually available generally make possible the closed ashpit system in the merchant marine, and thereby insure clean and comfortable fire rooms.

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[1] Machinery of the New Vessels of the U. S. Navy. George W. Melville. Transactions Society of Naval Architects and Marine Engineers. 1893.

This method of application also presents an opportunity which it shares with the induced system for utilizing the heat of the waste gases, which cannot be attained with the closed fire-room system. The manner of application will be illustrated in a succeeding chapter.

The practical results of the introduction of the closed ashpit system in place of natural draft are clearly shown in the record of four voyages of S. S. Dania under each of these conditions, which is here presented in Table No. 115. The total consumption of coal per day was reduced 13 per cent, while the time occupied in making the voyage was decreased nearly 5 per cent.

Table No. 115.— Saving by Forced Draft on S. S. Dania.

Conditions.	Days Steaming.	Knots per Hour.	Consumption of Coal per Day.	Consumption for all Purposes per Day Steaming.
Natural draft, four voyages	17.00	7.50	9.73	10.70
Forced draft, four voyages	16.21	7.58	7.76	9.31

Closed Fire-Room System.— This system of mechanical draft, in which a plenum condition is maintained in a practically air-tight fire room, is evidently impracticable for general boiler practice on land. Its existence is in fact largely due to the exigencies of modern naval warfare. It lends itself admirably to the necessities of vessels in actual warfare, for it is necessary that the openings down to the engine and boiler rooms should be kept as small as possible; and in all cases the machinery department would be closed down and the air supplied by artificial means during an engagement. Fans installed upon the closed fire-room principle can be easily arranged to ventilate the engine and fire rooms as well as to increase the combustion rate. They thus perform a double duty and avoid the use of a second set, were this arrangement inadmissible.

The conditions existing in the naval marine are, however, decidedly different from those in the merchant service. In the case of the warship its maximum steaming capacity is seldom demanded, and then only for a comparatively short time, as during an engagement. Consequently mechanical draft, as applied to vessels of this class, is mainly auxiliary in its character, but nevertheless a practical necessity. By its employment it is possible to construct the machinery within the limits of space and weight which are sufficient for ordinary service, while the reserve of power is stored in the light fans and fittings, instead of in the cumbrous boilers and machinery. In this fact is summarized one of the most important advantages of mechanical draft for marine purposes.

Under ordinary cruising conditions, when the stacks are of moderate height, the fans may not be required, but they must be of form, construction and capacity sufficient to meet at an instant's notice the maximum demand that may be made upon them. For instance, a vessel of the cruiser type, which may be required in case of necessity to develop 9,000 to 10,000 horse-power, may at the usual cruising speed of 10 or 12 knots require only 1,500 to 2,000 horse-power.

The conditions are well exemplified in Table No. 116,[1] which presents the results of a series of tests of U. S. S. Charleston at various rates of speed. The rapidly increasing horse-power with increased speed is to be noted, as is also the far more rapid increase in the power of the blowers, made necessary to meet the requirements. The relation between the actual efficiency, as shown in the coal per indicated horse-power and the knots per ton of coal, is of interest as indicating the difficulties in the way of obtaining even a slight increase of speed when it is well up to the maximum.

Table No. 116. — Results of Tests of U. S. S. Charleston.

ITEMS.	SPEED IN KNOTS.					
	13	14	15	16	17	18
I. H. P., main engines	2,220	2,820	3,550	4,370	5,220	6,120
Coal per I. H. P. per hour	2.2	2.1	2.0	1.9	2.1	2.5
I. H. P. of blowers	0.0	2.5	6.4	17.6	36.8	69.6
Knots speed per ton of coal	4.34	4.00	3.68	3.41	2.84	2.22

The increased steaming capacity per square foot of grate and per ton of boiler which results from the introduction of mechanical draft is shown in the case of the closed fire-room system by the results of trial performances of vessels of the British Navy, as presented in Table No. 117.[2] While in the later vessels there were improvements in the way of more economical engines, yet the greatest saving in weight, per indicated horse-power, was brought about by the introduction of mechanical draft. These tests, which were in the early days of its introduction in the naval service, exerted a strong influence on subsequent construction. Said Mr. Richard Sennett, in 1886:[3] "The only system that has yet had any extended practical trial is that of closed stokeholds worked under air pressure, which was described at considerable length by Mr. R. J. Butler in 1883.

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[1] Transactions Society of Naval Architects and Marine Engineers. 1893.

[2] Transactions Institution of Naval Architects. London, 1886.

[3] Closed Stokeholds. Richard Sennett. Transactions Institution of Naval Architects. London, 1886.

"Since that time all ships built for the Royal Navy, and many other vessels, have been fitted with this system, and the results obtained have been most satisfactory."

Table No. 117.—Results of Trial Performances of Similar Ships in British Navy.

Draft.	SHIP.	Date.	Steam Pressure, Pounds per Square Inch.	I. H. P.	Weight of Boilers. Tons.	Area of Fire Grate. Square Feet.	I. H. P. per	
							Square Foot of Grate.	Ton of Boiler.
Natural Draft, Open Fire Rooms.	Inflexible	1878	60	8,483	756	829	10.21	11.22
	Colossus	1883	64	7,492	594	645	11.62	12.61
	Phaeton	1884	90	5,588	462	546	10.23	12.1
Forced Draft, Closed Fire Rooms.	Howe	1885	90	11,725	632	756	15.54	18.5
	Rodney	1885	90	9,544	474	567	16.83	20.1
	Mersey	1885	110	6,628	306	399	16.61	21.7
	Scout	1885	120	3,370	174	207	16.28	19.3

The weight of boiler includes water, funnel, uptakes, fittings, spare gear, etc.
Steam blast was used in the case of the Colossus.

Among the advantages of the closed fire-room system is that of preventing all escape of flame and smoke to the fire room, the leakage being all inward to the fires. The objection is frequently brought against this system that its use is injurious to the boilers, tending to cause leakage at the tube plate when the doors are opened. This is generally considered to be due to the chilling action of the comparatively cold air, which thus rushes directly through the furnace chamber to the tubes. The leakage which sometimes occurs appears to be due to the buckling of the tube plate, whereby the joints between the tubes and tube plate are opened up, as a result of the unequal contraction under the chilling effect of the air.

The attempt has been to palliate rather than overcome this difficulty by the introduction of protecting ferrules at the tube ends. This tendency to leakage under heavy air pressure should not of necessity be charged against the method of producing the draft. "Because," as stated by the experienced marine-boiler builder, Mr. A. F. Yarrow,[1] "a certain boiler does not stand a given air pressure without the tubes leaking only proves that this air pressure is too much for that boiler, but does not prove that this air pressure is too much for every boiler, especially in face of the fact that locomotive engines are working all over the world at air pressures varying from 3 inches to 8 inches." There can be no question that the problem, when fairly considered, can be solved so as to avoid any trouble from this source. It can to a great degree be overcome by an arrangement of dampers and fire doors, such that the draft may be shut off while the doors are open.

Induced System. — This system, whereby a partial vacuum is produced within the furnace, is substantially the same in its effect as a chimney, which it most closely imitates in its action. It possesses, however, not only greater intensity, but other advantages that do not pertain to the chimney. In fact, it is the most natural method of applying mechanical draft, there is no change in arrangement of the boilers from that which obtains when a chimney is used, and there can be no question as to its adaptability when consideration is given to the high pressure differences and rates of combustion which are secured in a similar manner in locomotive practice.

It is, on the whole, better subject to control than the other systems; its leakage is always inward, avoiding inconvenience from flame and smoke at the fire doors, which, however, is only liable to occur under heavy air pressures and when proper damper arrangements are lacking. On shipboard it produces excellent ventilation with open fire rooms, thereby reducing their temperature. It is cleanly, lends itself readily to control by the dampers which may be introduced for the purpose, and can by simple means be rendered absolutely automatic, requiring no attention whatever from the fireman.

For all classes of work, except possibly on shipboard, in which the construction may prevent, it is well adapted, it usually being possible to place the fans above the boiler, or elsewhere overhead, in such a manner as to economize floor space.

The early objection to this system, before it had been extensively adopted, was that the fans could not stand the high temperature of the gases passing through them. The best refutation of this statement lies in the fact that large numbers of the Sturtevant fans have been running for years under these conditions, handling gases from 300° to 500° in temperature, and even higher. Of course these fans require to be of special construction to withstand the heat, and must be provided with means for keeping the bearings cool. But these features were long ago introduced by this Company, and today the decision between a forced or an induced system is to be made independently of the question of durability of the fans.

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[1] Boiler Construction Suitable for Withstanding the Strains of Forced Draft so far as it Affects the Leakage of Boiler Tubes. Transactions Institution of Naval Architects. London, 1891.

The inherent relative efficiencies of the forced and induced systems are difficult of determination because of the many modifying circumstances which exert their influence. Certain comparative tests have, however, been made, principally upon marine boilers; but they can hardly be taken as absolutely conclusive. Among them are those conducted at the Portsmouth (England) Dock Yard[1] in 1890, the results of which are summarized in Table No. 118. Both tests were made on the same boiler, the induced draft being dismantled and replaced by the forced draft. To the decreased tendency of induced draft to blow holes through, as compared with forced draft, and the better regulation and distribution of the air, may undoubtedly be attributed a part of its superiority. It is also claimed that the action of the air and gases upon entering the tube ends is such that with forced draft a contracted vein is formed which prevents close contact, while with induced draft the partial vacuum in the tubes causes the flow to be uniform across the section of the tube, and thereby insures better contact and readier abstraction of heat.

Table No. 118. — Results of Experiments at Portsmouth Dock Yard with Boilers of H. M. S. Polyphemus.

Kind of Draft.	Duration. Hours.	Average Steam Pressure. Pounds.	Temperature.		Total Coal Consumed. Pounds.	Total Water Evaporated. Pounds.	Lbs. Water Evaporated per Lb. Coal.		Pounds of Coal Consumed per Hour per Square Foot of Grate.	Lbs. Water Evaporated per Hour per Sq. Ft. Grate.		Approximate I. H. P.
			Feed.	Atmosphere.			At Actual Temperature.	At 212°.		At Actual Temperature.	At 212°.	
Induced,	96	74.2	62°	69.9°	80,600	777,044	9.64	11.13	40.4	389.6	450.4	426.
Forced,	96	77.3	51	49.8	94,500	759,338	8.03	9.3	47.3	381.	444.	395.

The induced system presents an opportunity for the introduction of air or water heaters which are to abstract the heat from the waste gases, thereby securing one of the greatest possible economies in the modern boiler plant. In fact, no plant of reasonable size should be equipped without a heat abstractor of some form in connection with the fan. And, on the other hand, no form of abstractor can be more efficiently installed than in connection with a fan which possesses the ability to readily overcome the increased resistance.

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[1] Transactions Institution of Naval Architects. London, 1895.

The reduction in temperature which thus results not only increases the efficiency of the plant, but has an appreciable effect upon the proportions of the fan; for upon the temperature of the air and gases which pass through the fan must depend its size and speed to accomplish the desired results in the way of draft production. In a fan operating under the plenum or forced system, the volume of air supplied to the fire is substantially the same as that delivered by the fan, making no allowance for leakage. But with induced draft the fan must handle a volume of air and gases which, although the same in weight, is greater in volume practically in proportion to its increase in absolute temperature. Disregarding leakage, the weight is greater than the air admitted by the portion of the coal which has entered into chemical combination with it. On a basis of 18 pounds of air per pound of coal, this additional weight amounts to 5.5 per cent, while with a supply of 24 pounds it is 4.2 per cent. This increased amount may enter into any refined calculations of fan capacity, but it is unnecessary to go into the refinement of making allowance for difference in specific gravity or for moisture in the air or fuel.

As the capacity of a fan, both in volume handled and pressure produced, is easily varied by a change in speed, sufficient accuracy is secured in ordinary design by considering that air is the fluid handled, and that the general relations presented in Table No. 113 will exist between volume, weight and pressure and the speed and power of a fan at different temperatures.

CHAPTER XI.

ADVANTAGES OF MECHANICAL DRAFT.

General Advantages. — In considering the advantages of mechanical draft, comparison is obviously to be made with the chimney. The merits of the mechanical method of draft production may be discussed, first, broadly, as to the general economic and convenient results, and, second, in detail, as to the particular points in which it excels chimney draft. In all cases the degree of superiority must, of course, depend upon the circumstances. The difference between land and marine requirements, for instance, forms an important factor. In the former, the practical abolishment of the chimney, the introduction of air or water heaters and the more perfect combustion stand out prominent in the advantages. In the latter, the reduction in the weight of the boilers, which have always been among the heaviest parts of the machinery, together with the combined flexibility and economy which are inherent features of the system, are of great importance.

Some of the advantages of mechanical draft are thus summarized by Mr. Alfred Blechyden:[1] "First, it seems fairly well established that, if the boilers are well constructed and are provided with ample room to ensure circulation, their steaming power may without injury be increased to about 30 to 40 per cent over that obtained on natural draft for continuous working, and may be about doubled for short runs. Secondly, such augmentation is accompanied, in normal cases, by an increased consumption per indicated horse-power. But, thirdly, the same or even greater power being indicated, it may, with moderate assistance of forced draft, be developed with a smaller expenditure of fuel, the grates, etc., being properly proportioned. Fourthly, forced draft enables an inferior fuel to be used; and, fifthly, under certain conditions of weather, when with normal proportions of boiler it would be impossible to maintain steam with natural draft, the normal power may with forced draft be ensured. In particular cases any or all of these advantages may be a source of economy; and the first of them may render possible that which would otherwise be impracticable."

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[1] A Review of Marine Engineering during the Past Decade. Alfred Blechyden. Proceedings Institution of Mechanical Engineers. London, 1891.

"Artificial draft," so states Mr. W. S. Hutton,[1] "can be readily adjusted to effect the combustion of different kinds of fuel at different rates of combustion. It permits efficient combustion of fuel of inferior quality, and enables a steady supply of steam to be maintained, independent of climate and weather. It enables the supply of air to be properly distributed to the fuel in the furnace to effect economical combustion.

"The supply of air above the fuel can be readily adjusted to effect combustion of the gases evolved by the fuel, and the supply of air below the fuel can be regulated to effect the combustion of the solid portion of the fuel, and the movement of the hot gases can be readily controlled."

There is no valuable feature of the chimney that is not possessed by the fan to at least the same, and in many cases to a more marked, degree. The very features which, as shown in the preceding pages, are most conducive to economy are those which are incidental to the use of a fan for draft production. While the recent extensive introduction of induced draft in stationary practice has done much to emphasize the advantages of this system, the general superority of mechanical draft, properly applied, has long been recognized by those who have given careful consideration to the matter; but never has the entire matter been treated in its collective aspect, as it is here presented.

In any consideration of the substitution of other means of draft production for chimney, the steam or compressed air jet may be primarily admitted, although it has been shown that, from the standpoint of economy of operation, the fan must be substituted for them. Nearly forty years ago Mr. D. K. Clark[2] "testified to the advantage of a rapid, or rather intense draft, in perfecting combustion and extinguishing smoke," upon which Mr. C. Wye Williams[3] was led to remark: "But the difficulty lies in the obtaining of this 'intense draft.' . . . The absolute *command of draft* for the generation of the required quantity of steam, to enable the engines to work to their full power, being then so essential, it becomes a question whether *other means* than *the natural draft* should not be resorted to; since, independently of the uncertainty in the amount of draft, and the consequent irregularity in the working effect of the engines, the cost of sustaining that draft may be so much in excess of what an *artificial draft* would be." M. Peclèt also at this time investigated the subject, showed the low efficiency of the chimney as compared with a fan, and recommended the use of rotary fans, applied for exhausting on the induced system.

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[1] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

[2] Combustion of Coal and the Prevention of Smoke. C. Wye Williams. London, 1858.

[3] Ibid.

Rankine[1] bases his calculations of results with forced draft on an air supply of only 18 pounds of air per pound of coal, while those upon chimney draft are based upon 24 pounds, and then remarks that "with a forced draft there is less air required for dilution, consequently a higher temperature of the fire, consequently a more rapid conduction of heat through the heating surface, consequently a better economy of heat than there is with a chimney draft." So Mr. D. K. Clark[2] states that "the system of forced draft opens the way for increase of efficiency in facilitating the adoption of grates of diminished area in combination with acceleration of combustion."

Messrs. Mills and Rowan,[3] writing at considerable length in 1889, discuss the subject in part as follows: "The principles of what is now becoming well known under the name of 'forced combustion' have been repeatedly advocated during past years by those who have devoted thought and study to the subject. The position assumed by them — which is now finding favor amongst engineers — has been, in brief, that the air supply required for combustion in furnaces can be more economically furnished by mechanical power than by the action of chimneys; and that the mechanical method has other advantages, which enable it to be preferred to the one that is older, but more imperfect. One of these advantages is the higher temperature of combustion, which is equivalent, with a boiler of good design, to an increased evaporative power of boiler, or to increased evaporative effect for the fuel. Another advantage, which has not been fully realized in any plan as yet introduced in practical work, is that the rate of travel and escape of the flame and hot products of combustion is under control. It is thus possible to cool them more completely than can be done when chimney draft is used, and this means a saving of heat which would otherwise be uselessly dissipated.

"Mechanical or artificial draft thus presents to us a method of economically furnishing the air supply to furnaces and producing a more efficient combustion temperature, while it also renders possible further economies due to retarding the movement and escape of hot gases and to preliminary heating of the air supply by waste heat or otherwise."

Mr. W. S. Hutton[4] states that "the economy that may be obtained by combustion with forced draft in a steam boiler is due to the increased rate of combustion and the increased efficiency of the heating surfaces produced by it, resulting in increased boiler power. The increase of power obtained depends principally upon the quantity of air brought into intimate contact with the fuel in a given time, but the power of the boiler may generally be increased from 40 to 100 per cent by the application of well-arranged forced draft."

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[1] The Steam Engine and Other Prime Movers. W. J. M. Rankine. London, 1885.

[2] The Steam Engine. D. K. Clark. London, 1890.

[3] Chemical Technology. E. J. Mills and F. J. Rowan. Vol. I. London, 1889.

[4] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

Viewed from the standpoint of the economic results to be obtained by the introduction of air heaters — in this case the Marland apparatus — and the substitution of a blower for a chimney, Mr. J. C. Hoadley[1] states that "there can be no doubt that the heat to be returned to the furnace would several times exceed that necessary to make the power required to drive the exhausting fan, to the operation of which the final temperature of the gases presents no objection. . . . In the construction of new works the outlay for the Marland apparatus will be, or at least *may* be, largely offset by the saving in the cost of a chimney."

In its relation to marine service, Mr. J. W. C. Haldane[2] points out that "the employment of forced draft does not involve any new principles in chemistry, but enables us with greater certainty to carry out those conditions which chemistry shows are essential to produce perfect combustion, and thus obtain higher efficiency per pound of fuel than can be obtained by natural draft." As indicating conclusively the economy of forced draft in marine practice, Mr. Haldane presents data, here given in Table No. 119, from the more elaborate classification of Mr. Blechyden. From a commercial standpoint the decreased heating surface per horse-power is only second in importance to the increased effciency per pound of coal.

The Engineering Record,[3] editorially commenting upon the matter of mechanical draft, asserts that "there are important advantages possessed by mechanical draft which are sure to make a place for it and, we believe, bring about its

Table No. 119.—Comparison of Results under Natural and Forced Draft.

AVERAGES.	Number of Steamers.	Heating Surface.		I. H. P. per Square Foot of Grate.	Coal Burned per Square Foot of Grate per Hour, in Pounds.	Coal Burned per I. H. P. per Hour, in Pounds.
		Per I. H. P., in Square Feet.	Per Pound of Coal per Hour, in Square Feet.			
For all steamers . .	28	3.275	2.14	11.22	17.08	1.522
For natural draft . .	22	3.560	2.25	8.91	13.92	1.573
For forced draft . .	6	2.412	1.72	20.98	28.15	1.336

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[1] Warm-Blast Steam-Boiler Furnace. J. C. Hoadley. New York, 1886.

[2] Steamships and Their Machinery. J. W. C. Haldane. London and New York, 1893.

[3] The Engineering Record. New York, Jan. 6, 1894.

extended application. The flexibility of a draft controlled by power is a most valuable feature. By the momentary increase in the motive power applied to the fan almost any desired increase of capacity can be obtained, and sudden or unusual calls can be met by simply an extra turn of the throttle valve which controls the fan engine. Mechanical draft enables feed-water heaters or economizers to be used in the flue and the waste of escaping gases utilized. Indeed, these two appliances go hand in hand, either one requiring the other for the best practical efficiency. Mechanical draft has no objectionable features on account of expense of the plant in first cost. A boiler plant, fitted with economizer, fan and short smokestack, can be installed, it is said, with a smaller expenditure of money than a plant of equal capacity without an economizer and provided with a good brick chimney, say 125 feet high. The saving in fuel due to the operation of the economizer is thus nearly all net gain. The application of power for the production of draft is an innovation in steam-engineering practice which seems bound to become generally endorsed in the near future."

Much more might be quoted in the same tenor from other sources, but that which has been here presented certainly appears sufficient to prove that mechanical draft has the unqualified endorsement of those who have investigated its merits. Although its efficiency has long been recognized, its recent rapid progress is the best evidence that, along with higher steam pressures, more rapidly running engines and increased engine and boiler efficiency, mechanical draft must take its stand as a most important factor in the accomplishment of economic results.

The remainder of this chapter will be devoted to a summary of the specific advantages of mechanical draft. In the preceding chapters it has been the endeavor to show the principles upon which its success and superiority depend, to point out wherein its advantages lie, and to present conclusive evidence to substantiate all statements. Additional evidence is introduced in certain cases which follow, and the closing chapter presents numerous examples of the manner of application of the Sturtevant fans, together with the results which have been obtained in practice.

Evidently, this is a book with a purpose, and that purpose is, first and all-important, to convince the reader, by *facts* and *substantiated* claims, that under all ordinary conditions mechanical draft is preferable to chimney draft; and, second, that this Company is, through its extended experience and large manufacturing facilities, in a position to introduce the best apparatus in the most intelligent manner. To a careful consideration of the following claims the reader is therefore invited.

Necessity. — In its broadest sense, the necessity of mechanical draft is to be measured upon a commercial basis. If by its introduction greater economy in the first cost or running expense of a steam plant may be secured, or even if in its operation it is more convenient and adaptable to the conditions, it may reasonably be considered as a commercial necessity. But there are conditions under which the introduction of a chimney would be impractical, or it would be extremely difficult to secure the desired results without the aid of mechanical draft. Thus, for instance, the draft required might necessitate a chimney of excessive height and cost; it might be impossible to further increase the capacity of an old, or erect a new, chimney; or the stability of the ground might be such as to make it undesirable to erect a chimney thereon. Under these conditions mechanical draft becomes a veritable necessity.

On shipboard, "it is perhaps not too much to say that the very high speeds that have been recently obtained by several cruisers of moderate dimensions would have been impracticable without the application of forced draft."[1] In fact, forced draft in ships of war is now recognized as a necessity. This is particularly true in the case of small vessels of the torpedo-boat type, in which the excessively high rate of combustion and the intensity of draft are such as would demand a chimney of such excessive height as to be absolutely impracticable in a vessel of this description.

Adaptability. — The chimney requires certain fixed and practically unalterable conditions for its location and erection, and, once erected, is only to a limited extent adaptable to changes in the requirements. The fan, as employed for mechanical draft, may, however, be adapted to a great variety of conditions; for the character of its construction, usually of steel plate, makes it possible to build it to meet almost any conceivable requirements. If restricted space absolutely demands, it may be made of comparatively small size and provided with an engine of extra power to run it at high speed. It can be placed above the boilers or suspended overhead, thus occupying no valuable floor space, as is the case with the chimney; or it may be placed in any convenient location upon the floor, or even in an adjoining apartment, and connection made to or from the boilers by means of an underground or overhead duct. At the same time the size or shape may be made such as to best adapt it to the location. Under no circumstances are the foundations necessary for the proper installation of a mechanical-draft plant to be compared in expense with those which are required for a chimney.

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[1] Closed Stokeholds. Richard Sennett. Transactions Institution of Naval Architects. London, 1886.

If the conditions are changed the same fan may frequently be made to meet the requirements by a change of location, arrangement or motive power. Thus, while a chimney cannot without great expense be increased in its capacity, the mere lengthening of the cut-off of the engine, or possibly its exchange for one of larger size, is all that may be necessary to adapt a fan to the new conditions. This is particularly true of an increase in the boiler plant.

The peculiar adaptability of the fan to the assistance of a chimney is of great importance, for by its use many a plant weak in chimney power has been brought up to the requirements. It has likewise proved its great value in making possible the use of economizers or other heat abstractors both in old and new plants. Its instantaneous adaptability to a change in the demand for steam renders it of particular value wherever this is a characteristic of the plant.

While with the chimney the draft must always be produced by suction, in the case of the fan both forced and induced draft are available, according to the requirements of the case. The fan may be driven at exactly the speed adapted to the most economical operation of the boiler plant, and may be instantly and, if desired, automatically adapted to other conditions.

A fan may be temporarily employed for the production of draft in the case of a plant whose location is not fixed, or during alterations or removal. The erection of a chimney for temporary use is hardly to be thought of.

Controllability. — One of the most important of the advantages possessed by mechanical draft is the perfect control which may be maintained over its action. Such control, in the case of a chimney, rests principally in the operation of dampers which restrict the air flow, but with a given temperature do not affect the intensity or pressure of the draft. Proper control of a fan consists in so regulating its speed — usually with dampers constantly open — that not only are changes produced in the air volume moved, but the pressure is varied in such degree as to overcome the resistances. Where the rate of combustion is practically constant, such control can be secured to a reasonable extent by manipulation of the throttle valve. But for more variable conditions and for all close regulation the throttle valve on the engine is operated by an Automatic Draft Regulator, a device so constructed that the throttle is opened as the steam pressure falls, and *vice versa*. The result is almost absolute uniformity of pressure, as is shown by steam-pressure records presented upon succeeding pages.

With a chimney the intensity of the draft depends upon the intensity of the fire, and is, therefore, least when the fire is low, which is usually the time when it is most required. With the fan, on the other hand, it is possible to instantly produce practically the maximum draft under these conditions. The control which is thus maintained over mechanical draft is of special value when it is desired to start up quickly fires which have been banked. This feature is particularly applicable on shipboard, where steam may be brought almost instantly up to the blowing-off point from banked fires when the vessel casts off its moorings, thereby obviating the noise of blowing off previous to starting. In the naval marine the importance is more marked, for the vessels may, in the case of an anticipated engagement, be kept ready to respond at an instant's notice; all that is necessary to this response being the opening of the steam valves upon the fan engines.

Flexibility. — This advantage possessed by mechanical draft is closely related to controllability; yet it is an essential feature of this device, which is only to a limited extent possessed by the chimney. The chimney is, in its very stability, the opposite of flexible. True it is that the amount of air handled by a chimney can be restricted by dampers, but the intensity of its draft is fixed by its height and cannot be changed to suit the conditions. The fan, on the other hand, being under constant control, is thoroughly flexible as to both volume and intensity of draft. Accordingly, in an already established plant, the fan is always ready to respond to a change in the requirements. In electric railway and lighting service this feature is of the greatest value, as it makes possible instant response to sudden demands. Likewise, in marine service, particularly in the navy, flexibility in the draft apparatus keeps it constantly suited to the conditions, and always in a position to readjust itself. As stated by Chief Engineer William H. Shock,[1] "it can be readily adjusted for the combustion of different kinds of fuel and for widely different rates of combustion, so that a given boiler may be worked under greatly varying conditions." As already shown, the power of a boiler may "be increased from 40 to 100 per cent by the application of well-arranged forced draft." On the other hand, this feature of flexibility makes it possible in most cases to largely increase the size of a steam plant before exceeding the possible capacity of the mechanical draft apparatus.

The automatic action of the fan, whereby increased resistance tends to increase the speed to overcome that resistance, is a prominent feature of its flexibility, which is absolutely lacking in the chimney.

Independence of Climatic Conditions. — The influence of a change in the temperature of the external atmosphere upon the draft of a chimney has already been shown in Table No. 107. This influence is so great as to make a decided difference between the draft in summer and that in winter. On shipboard, in particular, the intensity of the wind also affects the draft to such an extent as to frequently cause considerable inconvenience. The influence of damp and muggy days is everywhere recognized in its effect of deadening the fires. All these results are likely to occur when the chimney is the sole reliance for draft production.

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[1] Steam Boilers. William H. Shock. New York, 1880.

But the fan always operates independently of climatic conditions; the draft can be made as strong in summer as in winter, and on a muggy day as on one that is bright and clear. This latter feature is of almost inestimable value in electric railway plants; for it is on just such days, perhaps made worse by snow and slush, that the greatest demand is made for power, while the electricity is most rapidly dissipated, and with a chimney the increased demand for steam is least readily met. The positive character of the fan, which is thus displayed, renders it available at all times and independent of all external influences. This feature is very clearly evidenced by practical experience, as expressed by Mr. Alfred Blechyden:[1] "Under certain conditions of weather, when with normal proportions of boiler it would be impossible to maintain steam for the ordinary speed with natural draft, the normal power may with forced draft be ensured."

Portability. — A brick chimney, once erected, is a fixture; and even a steel-plate stack is practically non-portable, considering the difficulty and expense of its removal. Accordingly, the only resort, in case of change of location, is to leave the chimney as the buildings are left. This holds true even if the boilers are only to be removed to another portion of the same works. The chimney must remain; it is only suited to certain conditions, and is practically valueless unless those conditions exist.

On the other hand, the fan and its accompanying engine, arranged for the mechanical production of draft, are always readily portable. They may be removed to another part of the works, or to a distant place, with almost equal facility, and not infrequently may be set up again under decidedly different conditions; while the chimney would have to be abandoned altogether under similar circumstances. The advantage of mechanical draft under such circumstances is well evidenced in the experience of this Company, the original chimney, built years ago, being left as a useless incumbrance when, through force of circumstances, the boilers were removed to another location. Mechanical draft was here applied and the gases discharged through a short stack, which in its relation to the useless chimney is graphically presented in the reproduction of a photograph in Chapter XIII.

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[1] A Review of Marine Engineering during the Past Decade. Alfred Blechyden. Proceedings Institution of Mechanical Engineers. London, 1891.

Salability. — Closely linked to and dependent upon the feature of portability is the facility with which a fan, previously employed for draft production, may be sold, not of course at its original price, but, particularly in the case of small fans complete in themselves, at a price that is worthy of consideration alongside of the continuing fixed charges upon a chimney that has been abandoned. For experimental or temporary purposes a fan may even be hired at a reasonable rate, and its employment may avoid an absolute expenditure for stack or chimney.

The feature of salability is of especial importance when a plant is to be altered — increased, for instance — and a new fan of different capacity is required. The fan is always an available asset. Certainly such a feature should enter into any consideration of the best method of draft production.

Efficiency of Fan vs. Chimney. — The fact that the chimney is an exceedingly inefficient device for moving air has already been made clear in Chapter IX., and it has also been shown that a fan requires so little power by comparison that, considered simply as the means by which air is to be moved, the fan possesses advantages far and above the chimney. As also there shown, the adoption of the fan renders available the greater part of the heat necessary to the operation of the chimney, and presents the greatest opportunity for increased efficiency which exists in modern boiler practice. In fact, in this one feature of efficiency of the fan is to be found the source of its great economic superiority. A still further increase in the efficiency of mechanical draft results from the utilization, for feed-water or other heating, of the exhaust steam from the fan engine, thereby practically extinguishing the item of cost of operation.

As previously stated, the amount of steam that must be supplied for the operation of the fan depends upon the character and size of the plant. Evidently, the greatest proportional supply will be required when small fans are run at high speed and under excessive air pressure. These are the conditions encountered in the forced-draft contract trials of naval vessels. Nevertheless, in the case of 12 vessels of the United States Navy,[1] of various types and sizes, most of them equipped with Sturtevant fans, the average indicated horse-power of the blower engines averaged only 1.2 per cent of the total indicated horse-power of the main engines under the conditions of full-power forced-draft contract trials. The corresponding average air pressure in the boiler rooms or ashpits was 2.1 inches, and the average rate of combustion, so far as given, was about 40 pounds of coal per square foot of grate per hour.

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[1] Machinery of the New Vessels of the United States Navy. George W. Melville. Transactions Society of Naval Architects and Marine Engineers. 1893.

Omission of Chimney. — While the absence of the chimney is the natural consequence of the introduction of mechanical draft, and therefore a source of economy, there are conditions under which its omission may be a direct advantage in itself, regardless of economic considerations. This is particularly true in the case of torpedo and similar small boats, as already indicated, where a stack of proper dimensions to produce the draft would be out of the question.

Increased Rate of Combustion. — Independently of the greater economy with high rates of combustion, mechanical draft stands as the only means by which the increased rate may be economically obtained. Coincidently the boiler capacity must of necessity be greater, provided the grate area is maintained. The expense or inconvenience of a chimney, to obtain rates above 20 or 25 pounds per square foot per hour, becomes so great as to practically preclude an increase. As observed by Mr. A. J. Durston,[1] "as long as draft was dependent on a funnel for its production a much greater combustion than 25 pounds of coal per square foot of grate was rarely achieved; with artificial draft, on the other hand, the rate of combustion may be accelerated to any amount, and as a boiler's capability of transmitting heat without injury to itself is simply a matter of degree, experience has been necessary to determine the rates of combustion that can with safety be employed with different types of boilers." When it is considered that in boilers of the marine type the combustion rate resulting from the employment of mechanical draft is now carried as high as 40 to 50 pounds, that in torpedo-boat and similar service a rate of 70 to 80 pounds is frequent, and in locomotive practice as high as 120 pounds is not at all unusual, the possibilities of increased rates of combustion with mechanical draft are evident.

When the capacity of a boiler can be increased from 40 to 100 per cent by the application of mechanical draft, with the consequent higher combustion rate as already shown, there can be no question as to its desirability.

Efficiency of Combustion. — It has already been clearly demonstrated in Chapter VII. that combustion may be more economically accomplished with high than with low rates of combustion. For this reason it is common practice, in the introduction of mechanical draft, to so reduce the grate area as to decidedly increase the rate per square foot necessary to maintain the total rate previously existing with the larger grate. The thicker fires required, the better utilization of the air supplied, the higher temperature and the ability of mechanical draft to create the pressure difference necessary to these results have already been indicated as the elements of economy in higher combustion rates.

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[1] Some Notes on the History, Progress and Recent Practice in Marine Engineering. A. J. Durston. Transactions Institution of Naval Architects. London, 1892.

This fact is recognized by Chief Engineer William H. Shock[1] in his statement that "artificial draft has the great advantage that, all things considered, it is cheaper than natural draft for high rates of combustion." Hutton,[2] basing his statement upon the experiments of Spence, also asserts that "by the use of moderate forced draft a higher efficiency of combustion is obtainable than by using natural draft only." But efficiency is not to be measured alone by the economic combustion of the fuel; it must also include the commercial efficiency of the entire plant. If, therefore, as is evident, the capacity of a given plant may be greatly increased by the introduction of mechanical draft and its accessories, the fixed charges for a given evaporation will be decreased and the aggregate efficiency will be raised. The importance of an increased combustion rate in the accomplishment of such efficiency is well presented by Mr. F. Gross,[3] who makes the statement that "the special advantage is, to give the same economy when burning at twice the rate in half the number of boilers, or to make the steam as economically as is now done with natural draft and plain tubes when burning three times the rate in one-third the number of natural-draft boilers."

Burning Cheap Fuels. — The ability to utilize cheap fuels, which is an inherent advantage of mechanical draft, has already been pointed out in Chapter V. It was there shown that in a certain plant of 1,005 horse-power the introduction of the Sturtevant mechanical draft plant had resulted in a weekly saving of $126.00 in the fuel account. Further instances of economy resulting from the production of draft by mechanical means are presented in Chapter XIII. The intensity of draft required in the combustion of the fine refuse anthracites, or of tan, bagasse and the like, makes this method of draft production a practical necessity. Therefore, the economy incident to the utilization of such fuels must be considered as a direct result of mechanical draft. Its efficiency is, consequently, to be measured by the percentage of saving in the cost of fuel for a given evaporation, other things being equal; and this saving is always dependent upon the relative costs of the fuels in the given locality and the expense of handling them. But it is evident, from the results shown in Chapter V., that large savings may be assured by the adoption of mechanical draft and the utilization of cheap fuel in any locality where there is a reasonable opportunity to choose between the various kinds of fuel.

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[1] Steam Boilers. William H. Shock. New York, 1880.

[2] Steam-Boiler Construction. Walter S. Hutton. London, 1891.

[3] Recent Experience with Cylindrical Boilers and the "Ellis & Eaves" Suction Draft. F. Gross. Transactions Institution of Naval Architects. London, 1895.

The importance of mechanical draft in the combustion of cheap fuels is thus indicated by Mr. William Parker:[1] "Another question which forced draft has satisfactorily solved is the use of small and inferior coal. There is at least one British, and there are three Italian steamers now running, with very satisfactory results, burning nothing but very inferior coal, which could not possibly be burned with natural draft with the ordinary fire bars."

Economy in Quantity of Fuel. — There may be conditions under which a reduction in the amount of fuel consumed may be of greater importance than that of direct decrease in the total cost of the fuel, which, with lower-priced fuel, must usually be used in greater quantity. The question becomes most vital in its relation to marine practice. For short voyages a decrease in the total cost of the fuel, even if its quantity be greater, will usually result in a net gain. But when the voyage is of any considerable length the best coal is usually found to be most economical; no matter what the fuel may be, any reduction in its quantity resulting from the use of mechanical draft must add directly to the freight-earning dead-weight capability of the steamer. Thus, suppose that for a steamer of 2,500 tons dead-weight capacity 300 tons of coal are required for an Atlantic passage, with ordinary draft. A saving of 20 per cent of fuel, obtained by the adoption of mechanical draft, would mean an increase in possible weight of cargo of about 60 tons. A steamer with compound engines and mechanical draft would, therefore, be on a par with one having triple expansion engines and natural draft.

Incidental to any reduction in the amount of fuel, whether on land or sea, is the direct saving in the cost of transportation, handling and firing, and in the lessened amount of ashes. Independent of its character, the introduction of economizers or heat abstractors, made possible by the adoption of mechanical draft, presents one of the best opportunities for reducing the quantity of fuel.

Mechanical Stokers. — As stated in Chapter VI., the mechanical stoker is rendered most efficient under the steadiness and intensity of draft which results from the substitution of a fan for a chimney. Not only can the draft be easily regulated to meet changes in the rate of stoking, but it is capable of responding even more quickly than the stoker to a demand for increased evaporation. The admission of air through hollow grate bars, which is a feature of certain mechanical stokers, can only be successfully attained by the use of positive means like the fan. The results presented in Table No. 82 demonstrate the possible economy with a combination of mechanical draft and mechanical stoking.

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[1] On the Progress and Development of Marine Engineering. William Parker. Transactions Institution of Naval Architects. London, 1887.

Smoke Prevention. — The prevention of smoke has been a purpose sufficient in itself to often warrant the introduction of a fan to insure rapid combustion and the proper supply of the oxygen necessary to accomplish perfect combustion. The importance of the fan in this connection has already been shown in Chapter V. Although usually a mere incident to its application, yet the prevention of smoke by this means is in many cases one of the most important advantages of mechanical draft, to be measured not so much in dollars and cents as in the privilege of continuing the use of boilers in a community where the smoke-prevention laws are enforced.

Utilization of Waste Heat in Gases.— With the chimney a comparatively high temperature of the rejected gases is an absolute necessity to the production of the draft. The production of draft by means of a fan is, on the other hand, independent of the temperature of the gases, and there is, therefore, presented the opportunity to utilize the heat which is a positive and unavoidable loss in the case of a chimney. How great this loss usually is has already been shown in Chapter VI., and the importance of air and water heaters has been indicated. So far as the production of draft is concerned, the gases may be cooled down to atmospheric temperature, but the practical limit is necessarily above this because of the expense of the abstracting apparatus required.

The saving in fuel which may be accomplished under working conditions by introducing mechanical draft and economizers is evidenced in Table No. 120,

Table No. 120. — Results of Tests of Mechanical Draft Plants and Economizers.

Plants Tested.	Temperatures.					Fuel Saving.
	Gases Entering Economizer.	Gases Leaving Economizer.	Water Entering Economizer.	Water Leaving Economizer.	Gain in Temperature of Water.	Per cent.
1	610°	340°	110°	287°	177°	16.7
2	505	212	84	276	192	17.1
3	550	205	185	305	120	11.7
4	522	320	155	300	145	13.8
5	505	320	190	300	110	10.7
6	465	250	180	295	115	11.2
7	490	290	165	280	115	11.0
8	495	190	155	320	165	15.5
9	595	299	130	311	181	16.8

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[1] Mechanical Draft. W. R. Roney. Transactions American Society of Mechanical Engineers, Vol. XV.

which presents the results[1] of tests of nine plants, in nearly all of which the Sturtevant fans were employed. Under all ordinary circumstances an economizer can be relied upon to bring about a saving of from 10 to 20 per cent.

The intensity of mechanical draft also makes possible the introduction of such heat abstractors as the Serve tubes and retarders, the resistances presented by which would ordinarily prevent their introduction.

Economy in Space. — A chimney always requires specific ground area. Owing to its weight the foundations are expensive, and where the ground is not stable the expense may be almost prohibitive. The fan and its engine are comparatively light and require but little space; and even that may usually be taken where least valuable, as overhead, so that no direct charge can be made for ground area occupied. This is of great importance where land is valuable, and the saving may go far towards paying for the apparatus.

The special advantage of mechanical draft in this particular has already been pointed out in Chapter VI. and illustrated in Figs. 1, 2 and 3, where it was shown that this item might become of considerable importance.

The economizers, or heat abstractors, may also be placed overhead within the height generally allowed for a boiler house. Even the size and cost of the boiler house itself may be reduced.

On shipboard the matter of space is of the greatest importance, for every foot saved leaves just so much more for coal and cargo. In a discussion of this matter in its relation to the equipment of a transatlantic liner, Mr. James Howden,[1] with his system of forced draft, estimates the following items of saving and increased return (here reduced to round numbers in U. S. coinage) resulting from the decreased space in a given ship where forced draft is used: —

On Round Voyage.

Cargo, weight and measurement, 1,600 tons . .	$20,500
160 first and second class passengers, less cost of food,	8,000
80 third-class passengers, less cost of food . .	1,200
Total for one round voyage	$29,700
Total for ten round voyages	$297,000

Economy in First Cost. — As between the greater cost of a chimney and its foundations, and that of a fan with its engine and foundations to fulfil the same requirements, there can scarcely be a question. In Chapter VI. it was shown that the cost of the mechanical draft plant, in the instance quoted, was only 38 per cent of the chimney and damper regulators. In fact, the saving there indicated was just about sufficient to cover the cost of the economizers, which might, therefore, have been installed as a part of the mechanical draft plant without increasing it above that of the plant with chimney and without economizers. Consequently, the saving due to the introduction of the economizers might all be credited to mechanical draft. The further reduction in first cost, resulting from decreased size of boiler plant for the same output and from reduction of space occupied and building required, was shown to be sufficient to make the aggregate saving about three and a half times the amount expended for the mechanical draft plant.

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[1] On Forced Combustion in Furnaces of Steam Boilers. Transactions Institution of Naval Architects. London, 1886.

The following editorial comment,[1] as relating to marine practice, is pertinent and conclusive: "There appears to be no reason why, in all ordinary marine boilers, forced draft should not be applied with such good results as to insure, in one year, the recouping of the capital expended upon the apparatus and necessary alterations."

In a circular on the comparative cost, efficiency and earning power of a vessel fitted with natural draft, and one equipped with a forced-draft system by which the air delivered to the boilers by means of fans was specially heated and distributed, Mr. James Howden states that for a vessel of 500 feet length, 57 feet breadth and 38 feet depth, "the estimated reduction in the cost of the 6 forced-draft boilers having 36 furnaces, less than the 9 larger natural-draft boilers, with the cost of the additional mountings, uptake, funnel, steam and water copper pipes, flooring, boiler room and extra shipwork required for the latter, after deducting forced-draft fittings, is £13,000 [about $63,000]."

Of course, in all cases, whether land or marine, the relative cost of the two methods of draft production must depend upon, and be largely affected by, the existing circumstances. But the conditions must be exceptionally unfavorable when the mechanical draft plant cannot show decided economy in first cost when compared with the chimney.

Decreased Size of Boiler Plant for Given Output. — If, as shown, the output of a boiler may be increased by the adoption of mechanical draft, it must be conversely true that its size may be reduced and the initial output still maintained. In stationary practice this reduction in the size of a plant necessary to produce a given result decreases both the floor area occupied and the cost of the plant. This has been discussed in Chapter VI., in the consideration of the influence of mechanical draft on the ultimate efficiency of steam boilers. In marine service a still further gain results, for the space and weight saved may be turned to direct commercial account in the carrying of additional freight or fuel. The greater the amount of coal which can be carried without exceeding a given tonnage, the longer the voyage the vessel can make at a given speed without replenishing its coal. This applies particularly to naval vessels and to steamers with which quick passages are to be made; for it is apparent that the speed, and power required to produce the same, must bear such relation to the coal supply and the distance to be run that the coal shall be just sufficient. If it is sought to increase the efficiency of the steam plant by reducing the speed, this may be offset by the fact that, although the coal expenditure per day will be less, more days will be occupied in the run, and the total result may be practically the same; while in addition the vessel will have reduced its annual earning capacity.

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[1] Nautical Magazine, London, Vol. LVII., 1888.

Mr. Richard Sennett[1] distinctly points out that "the many advantages, as regards the power and speed of steamships, that are gained by this reduction in weight and space required for the boilers are too obvious to require enumeration; and it is, perhaps, not too much to say that the very high speeds that have been recently obtained by several cruisers of moderate dimensions would have been impracticable without the application of forced draft."

The greatly increased output per ton of boiler with mechanical draft is most forcibly presented by Table No. 117, wherein the average indicated horse-power shown per ton of boiler is 11.98 with natural and 20.6 with mechanical draft; that is, the total weight of the boilers is about 42 per cent less under the latter condition.

To such an extent as the dead-weight cargo-carrying capacity of merchant steamers is increased, so are their earnings enhanced; therefore, any means by which such results may be effectually accomplished demands the careful consideration of engineers. Referring to the Howden system of mechanical draft, Mr. J. W. C. Haldane[2] says: "For very many years it was considered that *moderate* combustion in good-sized boilers was more economical than forced firing in those of smaller dimensions. . . . Leaving theory out of sight and turning to actual performance, we find that by the employment of this system the boiler capacity for equal powers is from two-thirds to one-half of that required for natural draft; while its use secures much greater economy in fuel and decreased wear and tear. The reduced space thus required for boilers, and also their proportionately diminished weight, necessarily make this system vitally important for ocean steamers."

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[1] Closed Stokeholds. Richard Sennett. Transactions Institution of Naval Architects. London, 1886.

[2] Steamships and Their Machinery. J. W. C. Haldane. London and New York, 1893.

Economy in Operating Expense. — In fairness to both chimney draft and mechanical draft, the operating expense for a given boiler plant should comprehend interest, taxes, rent, insurance; depreciation and repairs upon the entire equipment, as well as fuel, labor and transportation charges, per unit of evaporation. It has already been shown that under all ordinary conditions a fan, with its engine, connections and short stack, costs less than a chimney to accomplish the same results; that a smaller boiler plant may with mechanical draft produce the same output as one of larger size with chimney draft; and that the cost of superficial area and of enclosing building will therefore be less. Hence the fixed charges will be less. If an economizer be introduced, it far more than pays for itself in the saving of heat accomplished. The actual steam expenditure for operating the fan will not in a plant of considerable size exceed 1 per cent of that produced, and even in a plant of small size will seldom if ever exceed 3 to 4 per cent. Ordinarily the exhaust thus produced will be utilized in heating feed water or buildings, and thus the absolute cost of operation be reduced to an infinitesimal amount. Even if this steam were wastefully thrown away, the economizer, if such formed a part of the plant, would compensate for this loss in addition to paying its own fixed charges. Such economy of fuel per unit of evaporation as may be the result of any arrangement of mechanical draft obviously reduces the total amount of fuel consumed and lessens the charges for labor and transportation. As a rule, any saving in fuel may be considered as a net gain, whether it be accomplished by an improvement in the economy of combustion of that previously used or by the substitution of a cheaper though less efficient fuel.

In this broad sense the economy of operating expense, including as it does all contingent items, must be the measure of commercial efficiency. The respective economies secured by various individual features of mechanical draft have already been pointed out in this chapter. Their aggregate effect is to conclusively prove that this method of draft production possesses economic advantages, not to mention immeasurable conveniences, which make it indispensable to the model steam plant.

Ventilation. — In stationary boiler plants the effect of mechanical draft upon the ventilation of the boiler room is of minor importance, but upon shipboard it is almost vital. Under the latter condition the furnishing of large volumes of air not only keeps fresh that within the fire rooms, but coincidently lowers its temperature, which is of almost as much importance. When the overpowering conditions of the ordinary marine fire room are considered, it is evident that such provision of fresh air as may result from mechanical draft is twofold in its effect. From a humanitarian standpoint, the men are kept in better physical condition, while in its mercenary aspect the results are directly evident in increased efficiency of service. As stated by Mr. Richard Sennett,[1] "when the system was first adopted it was suggested that the men might have some reluctance to work in closed stokeholds. This, however, has not proved to be the case, and the men have, from the first, worked as confidently and cheerfully as in ordinary stokeholds. The only effect that the closing of the stokeholds has had upon the men has been to enable them to do their work in more comfort in consequence of the better ventilation."

Summary of Advantages. — In the preceding discussion of the advantages of mechanical draft, the difficulty of presenting each as independent of the others must have been evident. To a great extent they are interdependent, and the possession of one advantage is evidence of the possession of others of similar character. In a brief summary, however, these may be more readily brought into accord. Thus the very adaptability of mechanical draft is indicative of the fact that it is more flexible than that produced by the chimney, is more readily controlled, and less influenced by climatic changes; while the apparatus for its production is more readily transported and has a higher potential value than a chimney. To a considerable extent these stand out as the conveniences of this method, regardless of their economies. When it is shown that increased efficiency can be secured by a method that is more convenient, the advantage of mechanical draft is established.

The actual omission of the chimney is sometimes of far greater importance than would at first appear, while the readiness with which the rate of combustion may be increased is doubly appreciated when it is shown that under proper conditions the efficiency of combustion may be increased thereby. The purely economic features are presented most prominently in the ability to utilize low-grade fuels, the resultant economy being shown in numerous examples here presented. The economy in the quantity of fuel consumed has, in its relation to the use of mechanical draft on shipboard, an advantage which is closely allied to that resulting from the decreased space occupied.

The economic results which may be secured through the introduction of mechanical stokers and devices for utilizing the waste heat of the gases are rendered most evident under the conditions of mechanical draft production, as are also the great advantage of preventing smoke and the blessings of good ventilation as they are exemplified on shipboard. The facts that the size of a boiler plant required for a given output can be reduced when a fan is substituted for a chimney, that the cost of the mechanical draft plant is usually far less than that of the chimney draft plant, and that its operating expense is likewise less under proper conditions, all point most conclusively to the purely economic advantages of the method which it is the purpose of this book to present. When these are considered in the light of the convenience and various other advantages of mechanical draft, its evident superiority to chimney draft must be conclusively established in the mind of any one who has read these pages.

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[1] Closed Stokeholds. Richard Sennett. Transactions Institution of Naval Architects. London, 1886.

CHAPTER XII.

THE STURTEVANT FANS FOR MECHANICAL DRAFT.

It has been made evident in the preceding chapters that the essential feature of mechanical draft is a fan blower or exhauster. It has been shown that the blowing engine and the positive rotary blast blower are not adaptable, that the steam or compressed-air jet is not economical, and that the disc or propeller form of fan wheel is not suitable for the purpose. The peripheral discharge type of fan therefore stands as the only form which it is desirable to employ for the production of draft. It is the purpose of this chapter to illustrate and describe this type of fan in all of its principal forms as built by this house. It is, however, manifestly impossible to present all of the multitudinous shapes in which these fans are constructed to suit the ever-varying requirements of different plants.

Prominent among the advantages of mechanical draft as displayed in the preceding chapter is that of adaptability, as is most clearly evidenced in this and the succeeding chapter. The steel-plate construction employed in all fans but those of smaller size lends itself most readily to perfect adaptation to the conditions existing in any specific case. The fan may, if absolutely necessary, be small and be operated at high speed, or, as should otherwise be the case, it may be large and run slowly. It may be constructed of steel plate in all sizes, and of cast iron in the different types of the smaller sizes. In the former material it may take almost any shape within the range of possible requirements, while either cast-iron or steel-plate fans are regularly constructed to discharge either horizontally at the top or bottom, or directly upward or downward. The pulley or engine, according as one or the other is employed, may be placed upon either side of the fan; while the engine may, to suit circumstances, be single or double, open or enclosed, with its cylinders above or below the shaft, or may be horizontal and of any required size. Or if desired a direct-connected electric motor may take the place of the engine in all but the largest sizes. The most important of these various arrangements are presented in the succeeding illustrations, while in Chapter XIII. are shown specific applications, in many of which the particular conditions demanded the construction and introduction of the particular forms of fans which there appear.

Steel Pressure Blower. — The type of fan presented in Fig. 15 is, as its name indicates, a pressure blower rather than a volume blower. That is, the wheel is of such dimensions as to make it possible to deliver a comparatively small amount of air under high pressure. This is the requirement in the case of cupola furnaces and forges for which this type was originally designed. Substantially the same requirement exists in the case of some of the under-feed mechanical stokers, where a very deep bed of fuel is maintained and considerable pressure is necessary to overcome its resistances, although the actual volume of air required is not great. Under similar conditions it is useful in connection with crematories, garbage destructors and the like. These fans are capable of producing a pressure as high as 20 ounces per square inch. The fan wheel consists of a light but strong hub with extending arms and a series of galvanized steel-plate blades or floats attached thereto. Conical side plates are attached to these blades and extend from inlet to circumference. The steel shaft, to which the hub is keyed, is supported upon either side by special continuous-oiling journal boxes, which are of such length and so thoroughly oiled that heating is practically impossible. The smaller fans are each provided with a single pulley, placed between the box and the fan case upon the right-hand side as one faces the outlet, the fan being then known as a *right-hand fan*. The larger-sized fans are each provided with two pulleys, one upon either side.

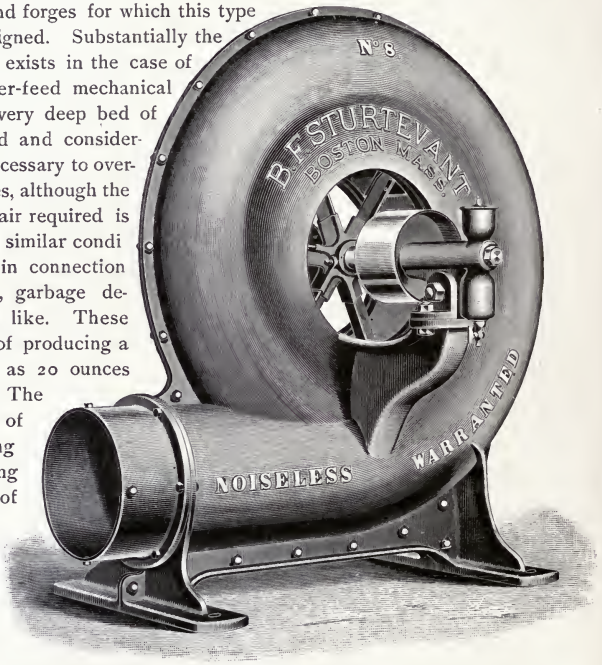

Fig. 15. Steel Pressure Blower.

"Monogram" Blower. — The "Monogram" Blower, illustrated in Fig. 16, and so designated because of the makers' monogram upon its side, is similar in general design to the steel pressure blower just described. The journal-box and shaft construction is the same, but the wheel is wider, being designed for the handling of a considerable volume of air; while the outlet for a given height of shell is larger, in order to accommodate the greater volume. Fans of this type are provided with only one pulley, which may be placed upon either the right or left hand side as one faces the outlet, thereby making the blower respectively right or left hand. The construction, which insures rigidity, is conducive to continuous operation at high speed, without any disagreeable noise or inconvenience from heating.

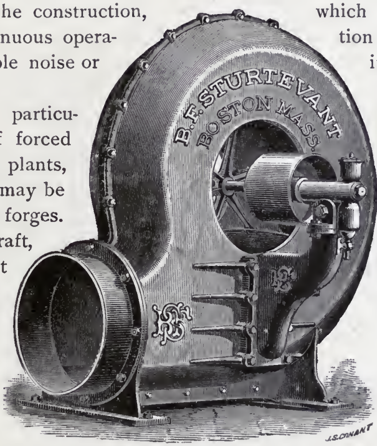

FIG. 16. "MONOGRAM" BLOWER.

These fans are particularly adapted for the production of forced draft in comparatively small boiler plants, and for creating such pressures as may be required for puddling furnaces and forges. The blower, as applied for boiler draft, may be located at a point where it is convenient to drive by belt, and thence the airpipe may be led to the ashpit. In fact, this is the type of fan first introduced by this Company, over a third of a century ago, for furnishing draft to steam boilers. Because of its ability to maintain considerable pressure this type has become of particular value in the production of draft for the burning of bagasse. It is also applicable in many cases where the steel pressure blower could otherwise be introduced and operated at moderate speed. This is true of its application for some types of mechanical stokers, where a pressure is desired somewhat in excess of that which it would be advantageous to maintain by a steel-plate fan. One of the extensive applications of the smaller sizes of these fans is for producing the blast required with the various forms of hollow grate bars, from which the air is discharged in minute streams directly into the bed of fuel. Exhausters of this type, having an inlet upon the side only, with both bearings and pulley outside the case, upon the other side, may be arranged with special cooling devices, so as to produce draft by the induced system, the gases being passed through the fan.

"Monogram" Blower on Adjustable Bed. — It is particularly important, in the case of a blower employed for draft production, that there should be no liability of its stoppage during working hours. So far as the construction of the Sturtevant blower is concerned, this is obviated by the character of the design and the perfection of the construction. But when driven by belt there is always a possibility of the tension thereon gradually decreasing until it suddenly becomes apparent in the slowing-down of the blower. To shut down long enough to take up the slack is a great inconvenience when boilers are depending upon the blower for the production of their draft. To avoid this necessity, the arrangement illustrated in Fig. 17 can be furnished. As is evident from the cut, the blower is placed upon a bed upon which it is adjustable, so that the belt may be continually kept tight. In its construction the bed consists of substantial steel side beams which are rigidly connected at their ends by castings, to which they are bolted. The blower itself is clamped to the beams by bolts passing down through its feet. The combination of beams and bolts serves to guide the blower and keep the belt aligned as it is drawn forward by means of the shackle bolt, which is attached to the front of the blower, just beneath the outlet, and passes through the front casting. To avoid interference with any connecting outlet pipe, resulting from the movement of the blower, a telescopic outlet is provided which is bolted to the end casting of the bed and within which a sheet-iron extension of the outlet of the blower slides as it is moved. The steel pressure-blower type of fan may be fitted up in the same way. The entire combination is readily portable.

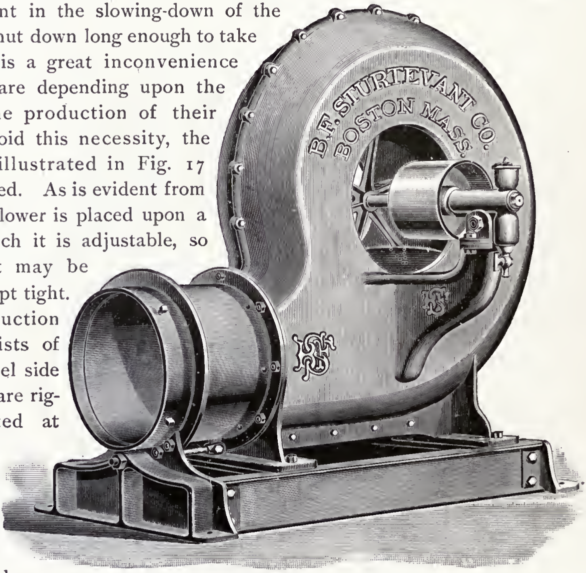
B.F. STURTEVANT CO.
BOSTON MASS.

Fig. 17. "Monogram" Blower on Adjustable Bed.

"Monogram" Blower on Adjustable Bed with Engine. — The arrangement just illustrated and described depends for its propulsion upon some means independent of the blower itself. But in such an important matter as the constant maintenance of draft it is particularly desirable that the blower should be provided with such means of operation as to render it entirely independent of any other source of power. In the case of a blower of the type under discussion, an engine may be readily combined with it upon the same bed. Such is the arrangement shown in Fig. 18, where the engine is double-cylindered, entirely enclosed, is capable of maintaining high speed, and subject to perfect regulation by the shaft governor. The enclosed feature of the engine prevents the throwing of oil and avoids the danger of injury to the bearings from the flying dust and dirt, almost always present in a boiler room. The same arrangement can be furnished with a single upright or a horizontal engine in place of the engine here shown. Any such combination makes it possible to start up the boilers independently of any other portion of the steam plant.

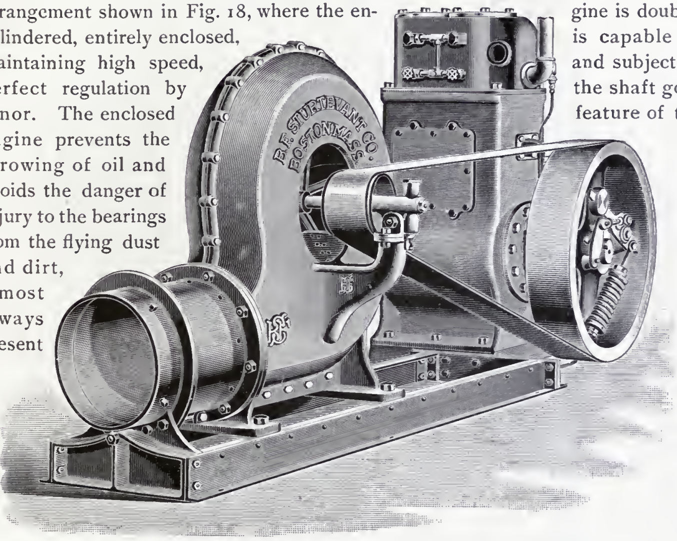
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FIG. 18. "MONOGRAM" BLOWER ON ADJUSTABLE BED WITH COMBINED DOUBLE ENCLOSED UPRIGHT ENGINE.

The bed proper is of the same general construction as that described in connection with Fig. 17. In case an engine is not desired, a counter-shaft (with tight and loose pulleys, if required) can be substituted, and the adjustable feature still retained. Evidently a pressure blower may be as readily fitted up in any of these various combinations.

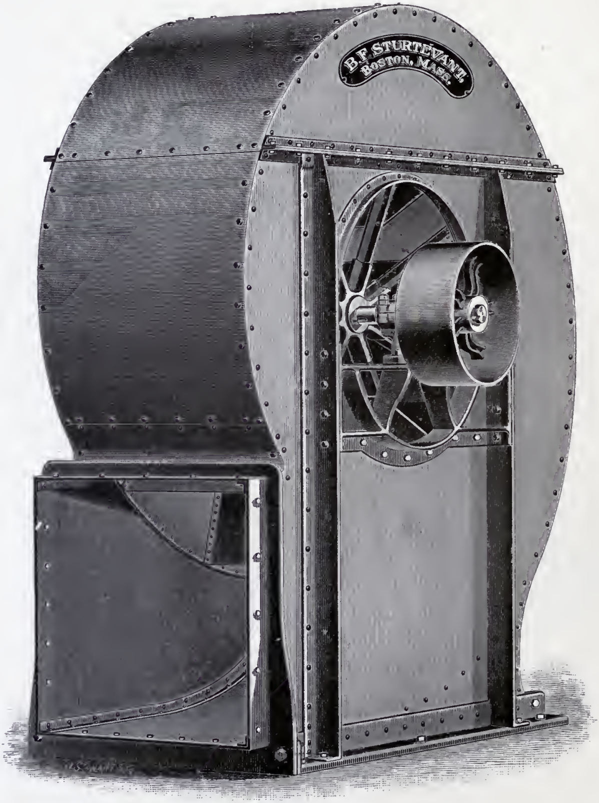

B. F. STURTEVANT,
BOSTON, MASS.

Fig. 19. Steel-Plate Blower with Overhung Pulley.

Steel-Plate Blower. — For moving large volumes of air under moderate pressure, the type of fan illustrated in Fig. 19 is extensively employed. The shell is constructed throughout of steel plate supported upon an angle-iron foundation frame and braced and stiffened by the same material. The entire construction is relatively light but strong, and may obviously be made to conform to any desired requirements. As here shown, the blower has a bottom horizontal discharge. This type is also regularly built to discharge horizontally at the top or directly upward or downward.

An inlet is provided in each side of the shell. A fan thus provided is designated as a *blower*, while one having only a single inlet (which is placed on the side farthest from the pulley) is known as an *exhauster*. This blower has a bearing in each inlet, with the fan wheel between, and the pulley overhung on the end of the shaft. The minimum of width is thus occupied, and this type of fan is thus rendered convenient for most applications for forced draft. In the larger sizes it is so constructed that the entire top may be readily taken off to obviate the objection to excessive height under the conditions of railroad transportation, which permit of only a certain maximum height.

The fan wheel which is enclosed within the shell is of the form illustrated in Fig. 20. It consists of a series of T-steel arms cast into the hubs — of which there are two — carrying the floats or blades, which, together with the side plates of the wheel, are constructed of steel plate. The fan wheel is carefully balanced to insure its steady running. The journal boxes, here clearly shown, are of the Sturtevant patent brush-oiler pattern, the oil being continuously fed to the bearing by means of a brush submerged in a reservoir which remains filled to a certain level.

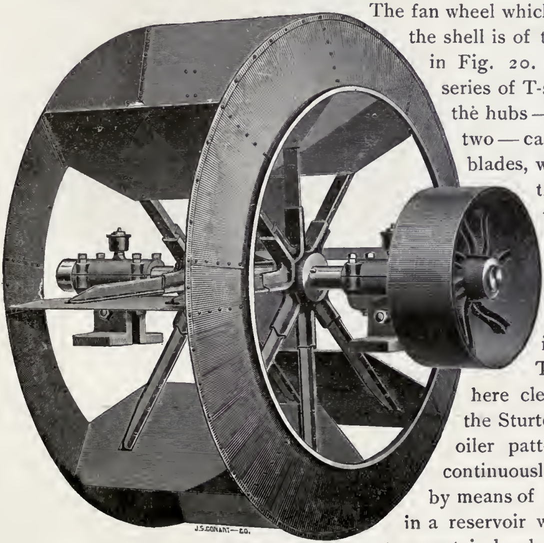

J.S.CONANT—CO.

Fig. 20. Fan Wheel.

Steel-Plate Blower on Adjustable Bed with Engine. — The adjustable arrangement which has already been illustrated in connection with the pressure and "Monogram" blowers is also applicable to the steel-plate blower, as is rendered evident in Fig. 21. A single engine is here shown, but a double engine or a counter-shaft could as readily form a part of the combination. The necessity of a belted engine is due to the high rotative speed of the fan, which would be excessive for a direct-connected engine of proper power. The utility of such an arrangement is obvious. It is readily portable, may be set up wherever desired without the preparation of special foundations, is not dependent for its operation upon any other source of power, and may be so regulated that the speed of the engine shall increase as the steam pressure falls. By this latter combination the range of variation in steam pressure may be reduced to a minimum — in fact, kept within one or two pounds.

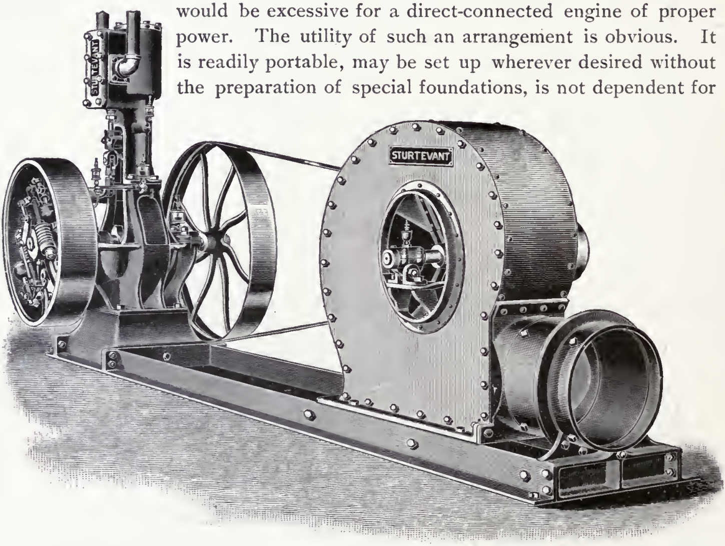


Fig. 21. Steel-Plate Blower on Adjustable Bed with Combined Single Upright Engine.

Evidently such an arrangement is suitable only for the production of draft by the forced method, but the entire equipment may, if desired, be placed on top of the boilers and the use of valuable floor space avoided; or it may, at the expense of comparatively little room, be placed along one side of the end boiler of a battery and discharge into an underground duct beneath or in front of the ashpits.

Steel-Plate Exhauster. — As already stated, the distinguishing feature of an exhauster is the single inlet, placed in the side farthest removed from the pulley or other means of propulsion. The standard form of steel-plate exhauster is shown in Fig. 22. This form of construction makes possible the ready connection of a pipe to this inlet for the purpose of exhausting air or gas from any particular space. The wheel being overhung upon the end of the shaft, and the pulley and boxes all being located upon the same side of the fan housing, the inlet is left entirely unobstructed and there is no opportunity for injury to the bearings by dust or heat. This type of fan is equally adaptable for use as a blower, all of the air then being taken in on one side. For the purpose of induced draft it is by far the best form, for a special Sturtevant water-cooled journal box may be easily substituted for the inner bearing and the transmission of heat along the shaft thereby prevented. If the air or gas handled is of excessively high temperature, the support may be set away from the shell by spacing pieces so as to allow a circulation of air between. This support, which may be rigidly bolted to the floor or foundation, carries the entire weight of the boxes, shaft, pulley and wheel, thereby removing from the fan casing all strain due to weight or tension of the belt. The casing itself is of substantial steel plate, except the outlet frame and bottom plate, which are of cast iron; while the wheel is of the same construction as that which has already been illustrated.

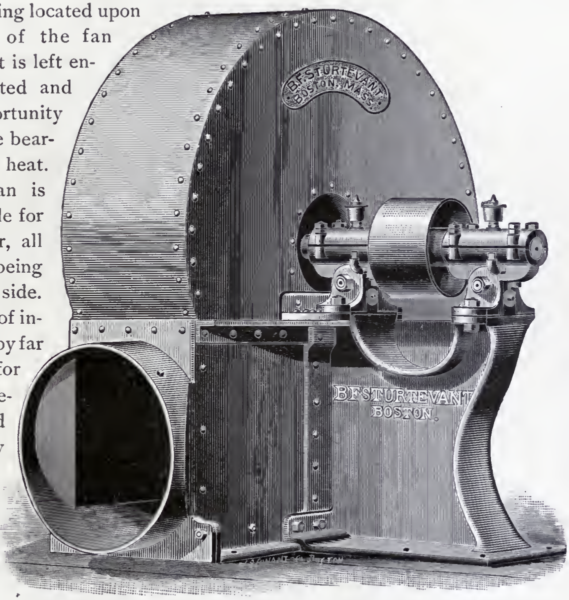


Fig. 22. Steel-Plate Exhauster with Overhung Wheel.

Steel-Plate Steam Fan. — It is always desirable that the means of propulsion for a fan should be as independent as possible of any other source of power; in other words, the motor adopted should be devoted solely to the driving of the fan. In the smaller sizes of fans of the pressure and "Monogram" types, the speed of rotation to produce the required pressure is such that a motor in the form of a steam engine directly connected to the fan shaft would be obliged to operate at too high a speed to remain durable; hence the belted arrangements which have already been shown. In the larger sizes of fans, however, particularly those of steel plate, the speed is such as to make direct connection practicable. A common form of this arrangement is that illustrated in Fig. 23. The fan itself is an exhauster, being identical in form and construction with that shown in Fig. 22, with the exception that the shape of the support is changed and that an engine is substituted for the journal boxes and pulley. This form of engine, which has its cylinder above the shaft, is of the same construction as the regular automatic upright engines built by this Company. The valve is of the balanced piston type, the cylinder is thoroughly lagged, the crank is accurately counter-balanced, and the crank pin is oiled from a stationary sight-feed oiler, attached to the frame of the engine. Large-cylindered low-pressure engines can be furnished in this type. Evidently this construction readily lends itself to application for mechanical draft, particularly under the induced system; for the wheel is overhung upon the end of the shaft and the inner journal may be water cooled.

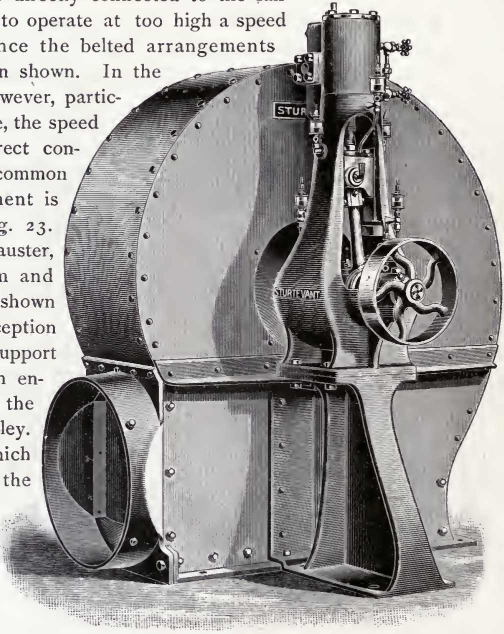
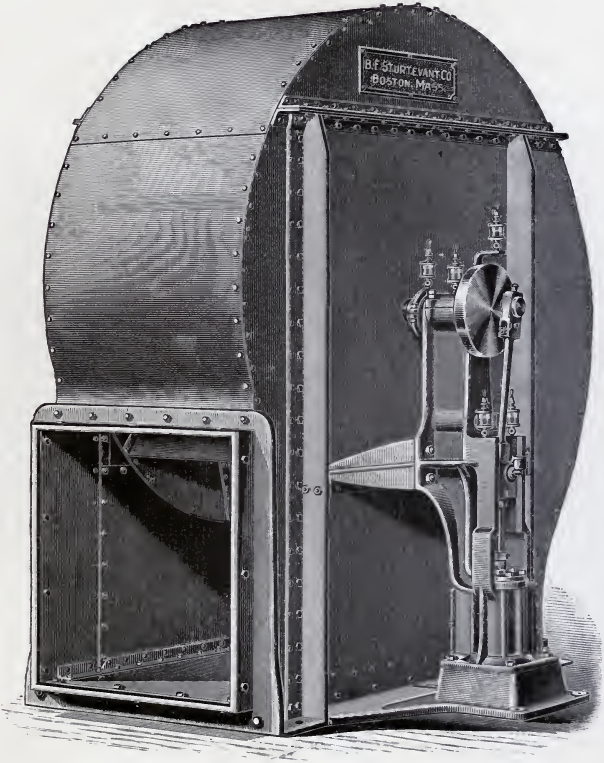


Fig. 23. Steel-Plate Steam Fan with Engine having Cylinder above the Shaft.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BOSTON, MASS.

FIG. 24. STEEL-PLATE STEAM FAN WITH ENGINE HAVING CYLINDER BENEATH THE SHAFT.

The form of steam fan illustrated in Fig. 24 is that employed in the larger sizes of full-housing steel-plate steam fans. As is evident, it is specially constructed for this particular use, its cylinder is beneath the shaft, and it possesses but a single bearing, the other bearing for the shaft being regularly placed upon a truss in the inlet upon the opposite side of the fan. When applied for induced draft, the shaft may be extended so that its supporting journal box can be placed outside the inlet connection. Both this bearing and that upon the engine may be chambered and kept cool by a constant circulation of water. The space which in the usual construction is left between the engine and the shell obviates any further trouble from direct transmission to the engine. Various applications of this and the previously illustrated form of steam fan will appear in the succeeding chapter. Both forms lend themselves to control by automatic draft regulators, which may be so arranged that as the steam pressure falls the engine speed and consequently the draft pressure and rate of combustion rise and more steam is at once generated.

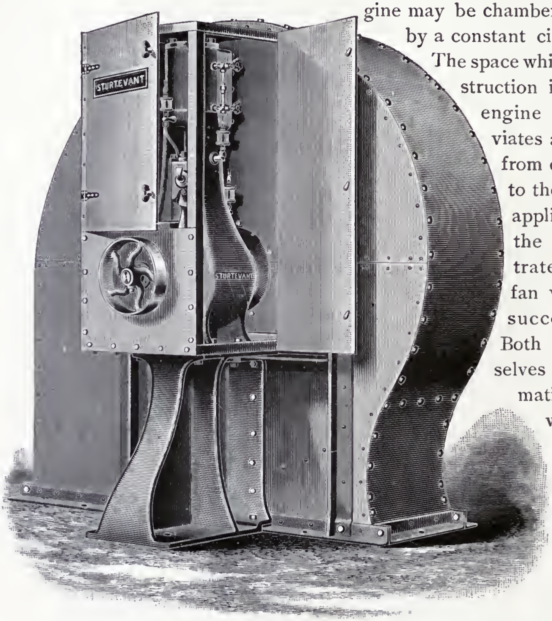

Fig. 25. Steel-Plate Steam Fan with Engine Enclosed.

Steel-Plate Steam Fan with Engine Enclosed.—The objection to the presence of an engine in some boiler rooms is usually that of liability to damage from the fine dust which is floating in the atmosphere and constantly tends to work into the bearings with disastrous effect. This objection may be removed by entirely enclosing the engine in a steel-plate casing as shown in Fig. 25. A regular form of double enclosed engine is shown in subsequent cuts.

Steel-Plate Exhauster with Inlet Connection. — When an exhaust fan is to be employed for induced draft it is frequently desirable to construct, in connection with and in fact as a part of the fan, an inlet connection in the manner indicated in Fig. 26. As there shown, with the shaft extended through the connection and supported by an outside journal box, the arrangement is particularly adaptable to any type of fan, whether steam or pulley, such as is shown in Figs. 19 and 24, in both of which the shaft is ordinarily supported by a bearing in the inlet. Naturally the external bearings would be provided with cooling devices if hot air or gas is to be handled, and therefore thereby rendered perfectly serviceable even under these somewhat trying conditions.

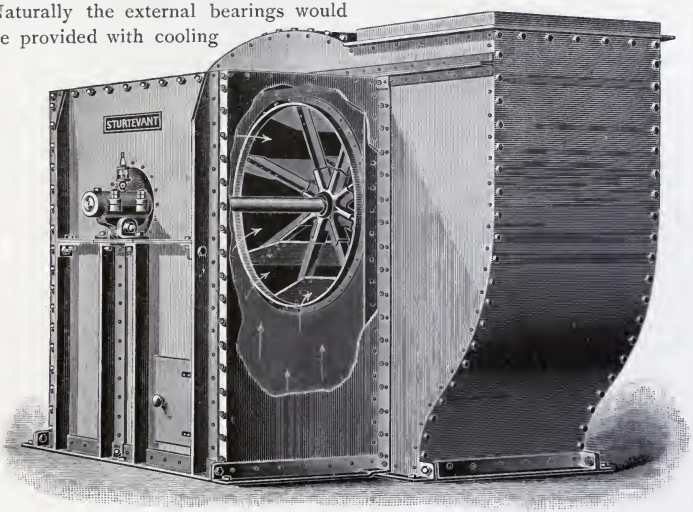

Fig. 26. Steel-Plate Exhauster with Inlet Connection.

As here represented the inlet connection is of steel plate, with angle-iron corner frames and additional bracing of heavy angle iron. It is provided with a door to permit of access to the interior of the connection and of the fan for the removal of soot and dust. Although shown with open bottom for the admission of air or gas, it may as readily be constructed so that the supply can be taken from above or through either side. The bottom connection is especially desirable if the fan is to be placed above the boilers and the gas taken from a flue beneath.

Special Steel-Plate Steam Fans.—The types of independent fans which have thus far been presented are those of regular form. But, in the adaptation of fans for mechanical draft, many special forms are required, particularly for application on shipboard. These are generally provided with independent engines, in each case directly connected to the fan shaft. Great variety in the character, form and proportions of these engines is necessary to make them readily adaptable; as a consequence, the differences between most of the fans, the illustrations of which here follow, lie fully as much in the engines, by means of which they are driven, as in the fans themselves.

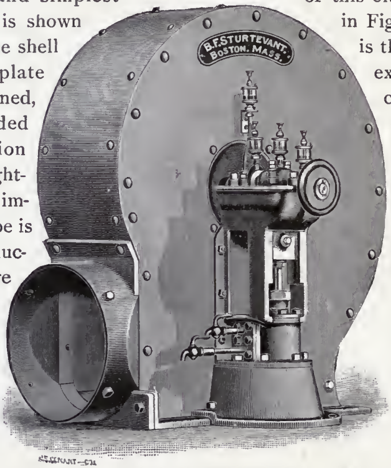
B. F. STURTEVANT, BOSTON, MASS.

Fig. 27. Special Steel-Plate Steam Fan, with Single Engine.

The smallest and simplest of this class manufactured by this Company is shown in Fig. 27. The general construction of the shell is the same as that of the regular steel-plate exhausters. The engine is self contained, carries the fan wheel upon its extended shaft, and is designed for operation at high speed. It is provided with sight-feed oilers and is adjustable in all important bearings. This size and type is serviceable for forced-draft production on small steam yachts where the grate service is limited in area, being used for the creation of under-grate pressure. Another type, also extensively employed for yacht and general marine work, is represented in Fig. 28. Here the foundation of the fan and engine consists of channel beams extending from the shell of the vessel to an interior support, thus bringing the top of the fan casing close up to the deck. Owing to the limited space in the steam yacht Sapphire, for which this was designed, the outlet was formed in the side of the casing, the air being deflected thereto by a curved plate within the casing. From this outlet a pipe leads to the boiler ashpit. Evidently this arrangement occupies the minimum of space. The engine is of the double-cylindered type subsequently illustrated in Fig. 32. It is particularly adapted for this location because of its compactness, its perfect balance, its ability to run at high speed for a long period and its enclosure from dust and dirt.

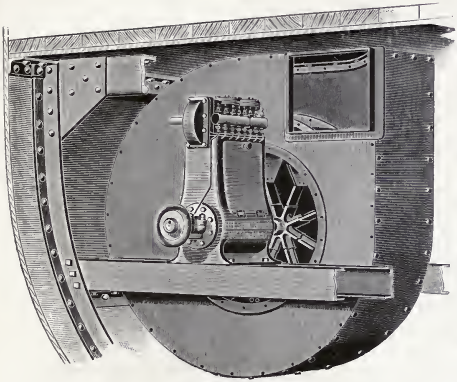

FIG. 28. SPECIAL STEEL-PLATE STEAM FAN WITH DOUBLE ENCLOSED ENGINE.

It is sometimes the case that an upright engine of the type just described will, if of adequate power, require a greater height than the conditions will admit. This difficulty may in some cases be avoided by using an engine of the same type with the cylinders below the shaft, as shown in a succeeding illustration; but when neither form is admissible resort must be had to a special type of horizontal engine. This was the condition which held in the design of the special fan shown in Fig. 29, which represents one of several fans constructed for U. S. S. Monadnock.

The engine is self-contained, having two bearings; and the fan wheel is supported on the end of the shaft. The crank and connecting mechanism are entirely enclosed, preventing the throwing of oil and the admission of dust. By the combined effect of the cast-iron bracket and the angle-iron sling, the engine is held rigidly in its place. Being carried close up to the deck, the fan being in fact fitted in between the deck beams, the least possible head room is occupied. Evidently such a fan can be arranged to deliver in any given direction or entirely around the circumference, as might be desirable in a closed fire room. Other forms for use in marine work are presented in the next chapter.

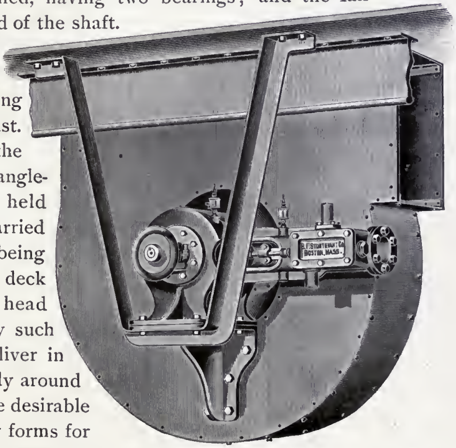

FIG. 29. SPECIAL STEEL-PLATE STEAM FAN WITH HORIZONTAL ENGINE.

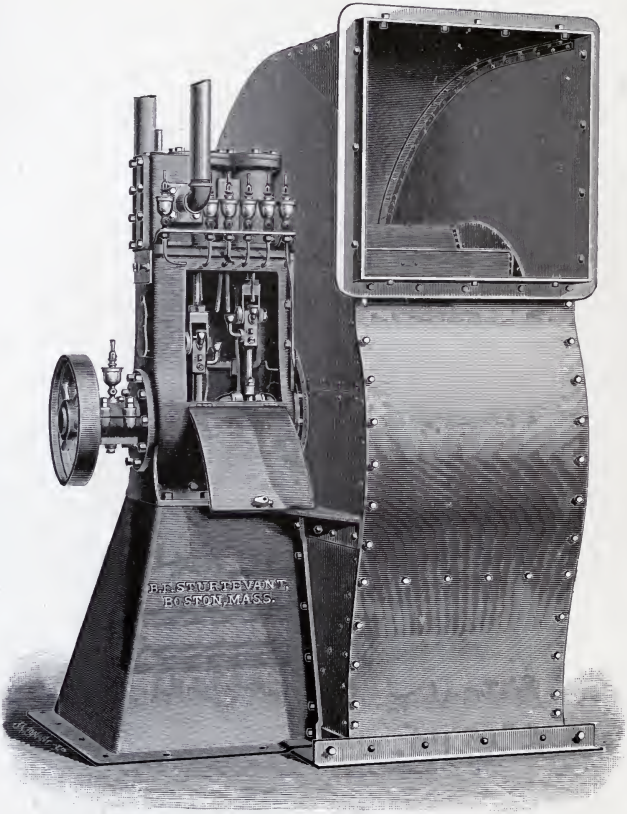
B. F. STURTEVANT,
BOSTON, MASS.

FIG. 30. SPECIAL STEEL-PLATE STEAM FAN WITH DOUBLE ENCLOSED ENGINE.

Still another form of the steel-plate steam fan with a double enclosed upright engine is shown in Fig. 30. This has a top horizontal discharge, and is applicable for either forced or induced draft. The engine is supported upon a substantial cast-iron base and carries the fan wheel upon its extended shaft. The hand wheel upon the outer end of the shaft is provided for starting the engine off the centre, when necessary. Large numbers of fans of this general type, but with the point of discharge to suit the conditions, have been furnished for the production of draft.

A pair of down-discharge fans is shown in Fig. 31, the combination with the engine forming a duplex steam fan, in which both fans are operated in unison by the same engine. A wheel is carried on each end of the shaft, which is provided with couplings between the engine bearings and those upon the fans, so that the engine can be entirely removed without disturbing the fans. By the arrangement for down discharge these fans may be placed above the boilers and the air delivered directly downward to them. If it be a stationary plant, a duct would connect each outlet to the boiler ashpits, but if used in the marine service either the closed ashpit or closed fire-room system of supply could be adopted. In the latter case the air would simply be delivered through openings in the deck corresponding to the outlets of the fans and thence discharged directly downward into the boiler rooms. The duplex feature reduces the height which it would be necessary to provide for a single fan of the same capacity.

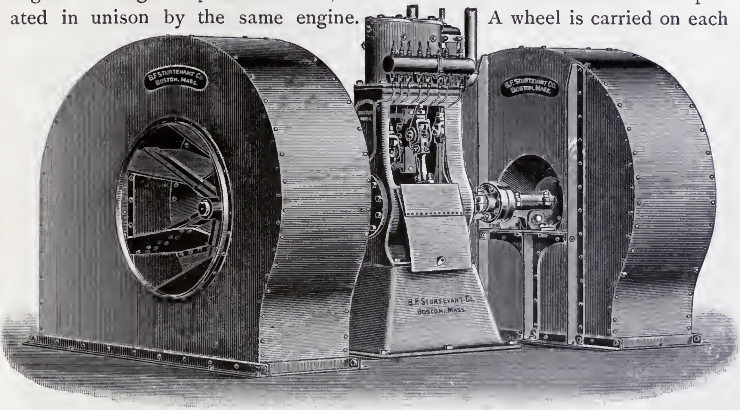
B.F. STURTEVANT CO. BOSTON, MASS.
B.F. STURTEVANT CO. BOSTON, MASS.
B.F. STURTEVANT CO. BOSTON, MASS.

FIG. 31. SPECIAL DUPLEX STEEL-PLATE STEAM FAN WITH DOUBLE ENCLOSED ENGINE.

Double Upright Enclosed Engine.—One of the first requisites of an engine for fan propulsion is the ability to operate continuously at high speed. The dependence which is placed upon the fan when it is utilized for mechanical draft is such that perfection in the engine is an important requisite. For moderate speeds and cleanly surroundings the types of single upright engines previously described effectually serve the purpose. But where the speed is excessive and the operation continuous, the engine should take the form represented in Fig. 32.

This type has two cylinders placed side by side in the same casting, the cranks are set opposite (at 180°), the reciprocating parts are thereby balanced and high speed is made possible. The cylinders are of large diameter as compared with the stroke, so that great power may be developed at high relative but moderate piston speed. The steam admission to both cylinders is regulated by a single balanced piston valve, and automatic relief valves prevent all danger of damage from water. All moving parts subject to friction are of steel, and the bearings are of ample size. Complete sight-oiling arrangements from a single oil tank connect with all of the bearings. The frame is so made as to entirely enclose all running parts, still leaving them accessible by merely opening the door.

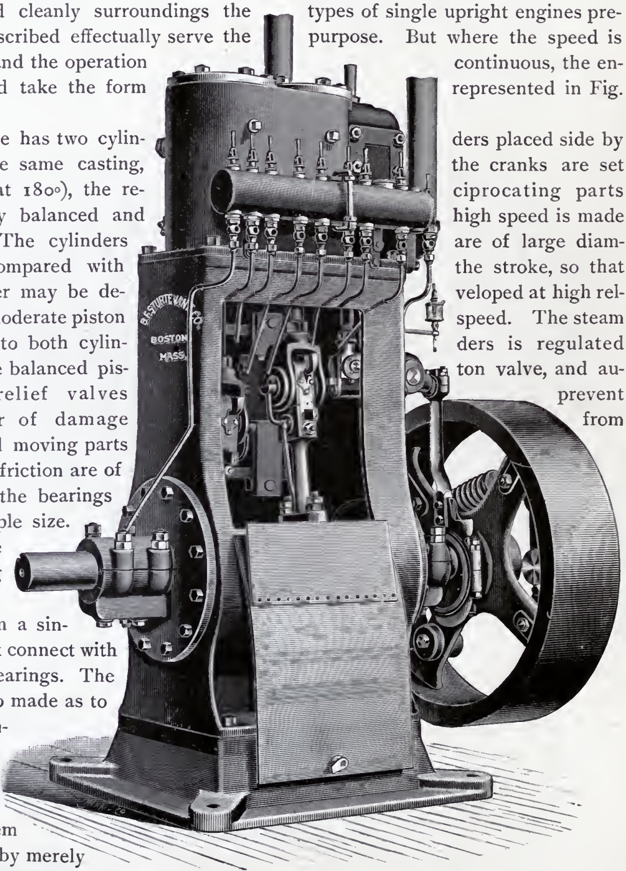

FIG. 32. DOUBLE UPRIGHT ENCLOSED ENGINE.

Special Steel-Plate Steam Fan, Double Enclosed Engine, Cylinders beneath the Shaft. — Insufficient height above the centre of the fan shaft frequently compels the substitution of an engine with cylinders beneath the shaft for one having its cylinders above the shaft. The same general type of engine is employed, however, the enclosed feature being retained, and the working parts, which are the same as those in the other type, being made readily accessible. This form of construction is clearly shown in Fig. 33. A single oil tank supplies all bearings through connecting tubes or wipers. The entire top, including the door and oil tank, may be easily removed if necessary. The automatic cylinder relief valves, which are plainly shown in the cut, form an important feature of this engine. The fan, which, like those previously described, is of steel plate, is designed to discharge directly downward through an outlet in the base at the end nearest the observer. Such an arrangement is particularly convenient for application under many of the conditions which exists in mechanical draft plants, where the air is to be forced into the ashpits.

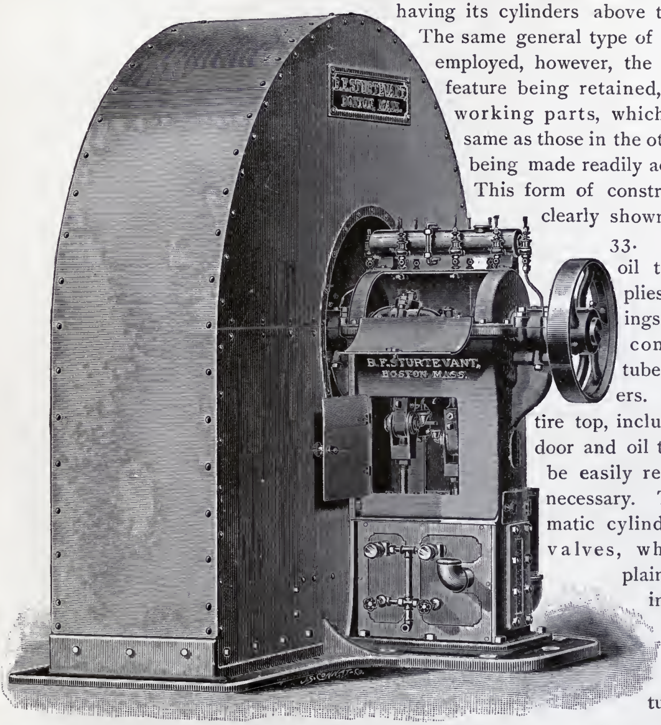
B.F. STURTEVANT
BOSTON, MASS.
B.F. STURTEVANT,
BOSTON, MASS.

Fig. 33. Special Steel-Plate Steam Fan with Double Enclosed Engine, Cylinders beneath the Shaft.

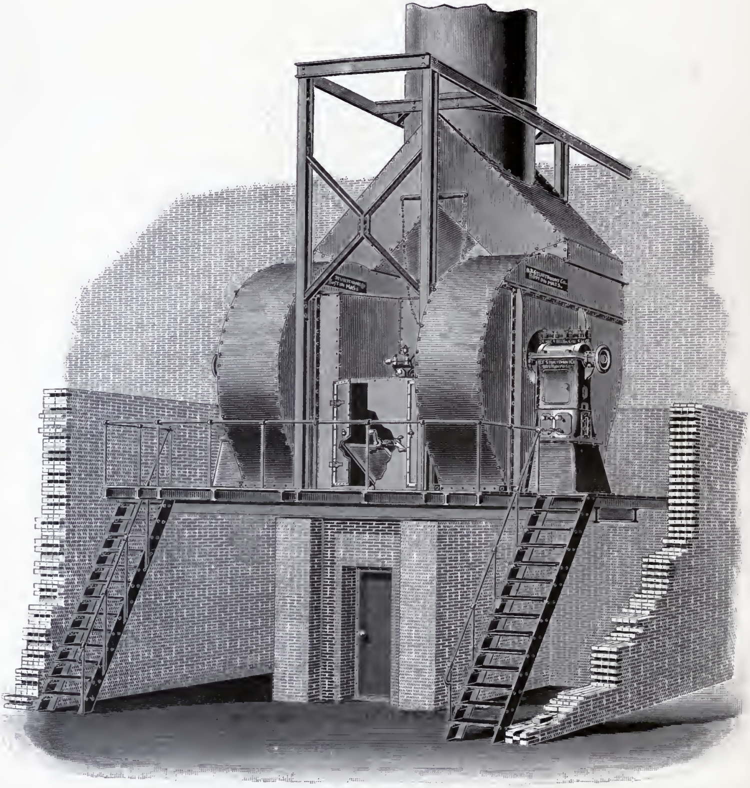
B.F. Sturtevant Co.
Boston Mass.

FIG. 34. SPECIAL DUPLEX STEEL-PLATE STEAM FAN APPLIED FOR INDUCED DRAFT.

Special Duplex Steel-Plate Steam Fan. — Two fans provided with engines of the type just described and set up in proper manner to form a duplex fan are shown in Fig. 34, which represents the arrangement designed for and installed at the Holyoke Street Railway Co.'s power house at Holyoke, Mass., for the production of draft by the induced system. The flue gases enter the brick chamber beneath the fans and thence pass to the sheet-iron connection between them. Here is located, as shown, a swinging damper, by means of which the gases may be made to enter either fan. Another damper, in the connection above the fans, likewise operates so that when one fan is in use the other is entirely shut off and is accessible for cleaning or other purposes. Either fan is capable of producing the maximum draft that is required by the entire plant. One fan may thus serve as a relay.

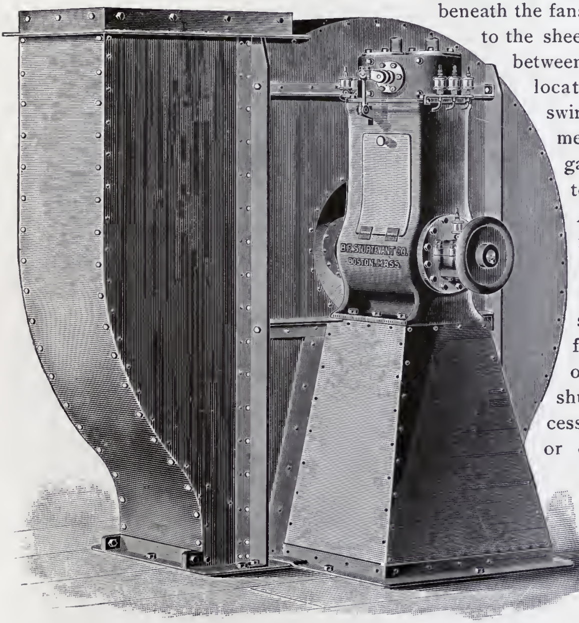

Fig. 35. Special Steel-Plate Steam Fan with Upright Compound Engine.

Special Steel-Plate Steam Fan with Upright Compound Engine. — The fan illustrated in Fig. 35 differs from those previously shown, in that it is provided with a special type of compound engine. This engine, because of its simplicity and economical performance, is of value where the exhaust steam cannot be utilized and high efficiency is important. A single oscillating valve performs the functions of the two valves necessary in the ordinary types, and with half the complication of moving parts.

Special Steel-Plate Steam Fan with Double Open-Type Engine. — Another type of double upright engine, not enclosed, and therefore suitable only for cleanly locations, is illustrated in Fig. 36. The relative size of this steam fan can be judged by the fact that the engine cylinders are 9 inches in diameter by 5½ inches stroke. For absolute rigidity, the engine was, in the plant from which this illustration was taken, set upon a special brick foundation. This renders its support entirely independent of the fan and removes all strain therefrom. The wheel, as in previous fans described, is overhung upon the end of the engine shaft, which is of large diameter and supported in three bearings within the base of the engine. The greatest care has been given to the continuous and effectual oiling of this engine, with the result that in numerous installations on transatlantic steamers it makes the passage without a stop. It is extremely compact, requiring the minimum of floor space for a given output, and is, therefore, especially valuable for use where but little space is available.

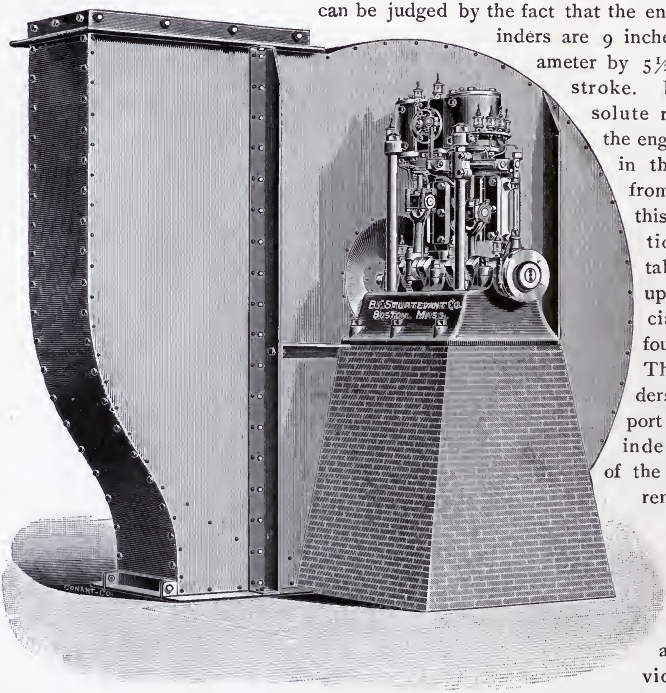
B.F. STURTEVANT CO.
BOSTON, MASS.
CONANT-CO.

Fig. 36. Special Steel-Plate Steam Fan with Double Open-Type Engine.

Special Cast-Iron Steam Fan with Double Horizontal Engine. — A somewhat unique form is presented in Fig. 37, which is from a photograph of one of the fans furnished for the U. S. S. Puritan. The side pieces of the shell are of cast iron, the rim being of heavy steel plate and the entire bottom of the casing being open for the delivery of the air directly downward. Supported upon projecting brackets cast on to one side of the shell is a double horizontal engine with its cranks set opposite so that the reciprocating parts are balanced. High rotative speed is thus made possible without objectionable vibration of the engine. The engine, in fact, consists of two engines, so constructed that either may be removed without disturbing the other. The valves, which are of the piston type, are actuated by eccentrics, transmitting the motion by means of rockers. A thorough system of sight-feed oilers, wipers and catch cups is provided, as is clearly shown.

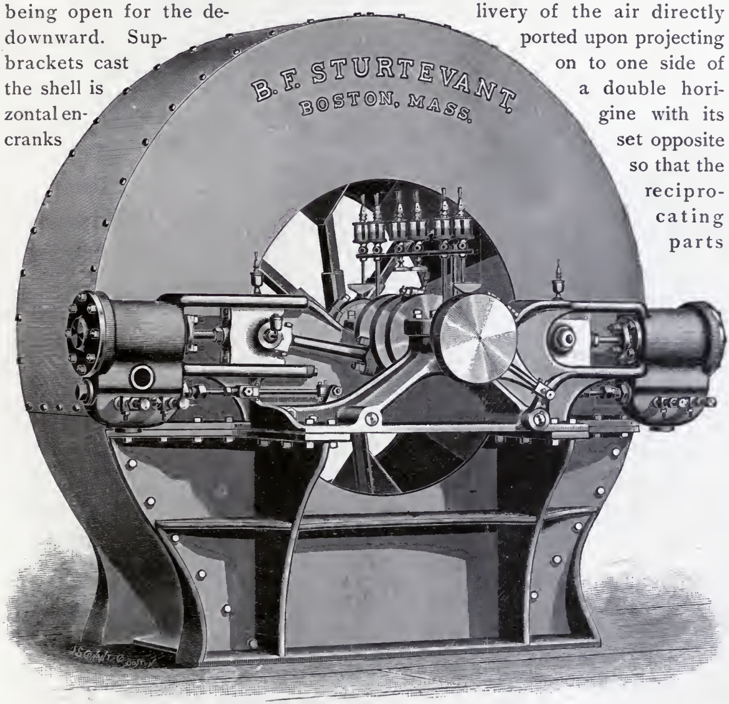

FIG. 37. SPECIAL CAST-IRON STEAM FAN WITH DOUBLE HORIZONTAL ENGINE.

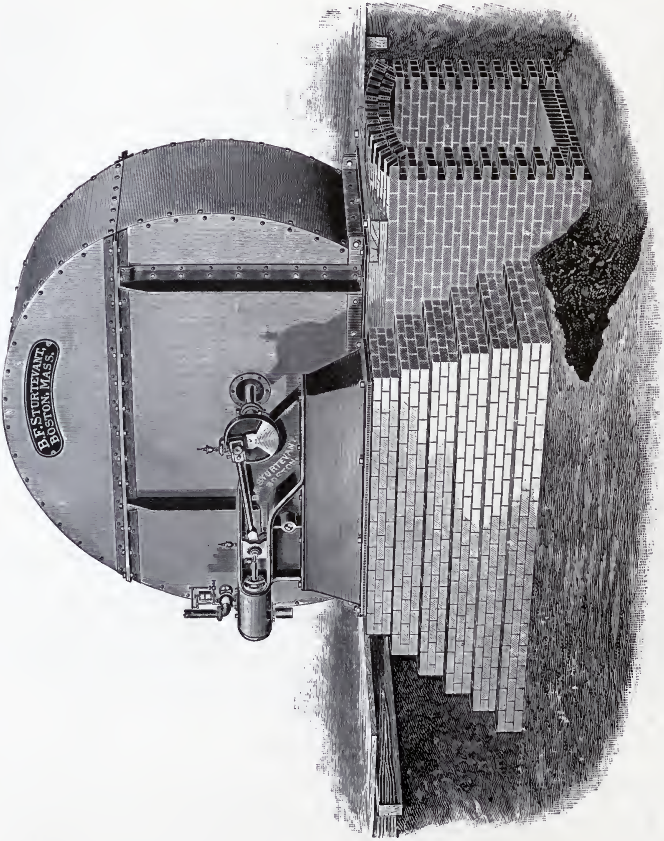
B.F. STURTEVANT,
BOSTON, MASS.
STURTEVANT
BOSTON

Fig. 38. Steel-Plate Steam Fan with Three-Quarter Housing and Horizontal Engine.

Steel-Plate Steam Fan with Three-Quarter Housing. — All the fans previously described have been of the full-housing type. Under certain conditions, such as a lack of available height or the desire to discharge into an underground duct, a fan having a portion of its scroll constructed in the brick foundation is both economical and convenient. The standard type of three-quarter-housing fan, as this form is called, is presented in Fig. 38, where it is shown as driven by a direct-connected horizontal engine, and arranged to deliver the air into an underground duct. Such an arrangement is of especial convenience for a large forced-draft plant, where the air is forced into the ashpits from a duct beneath or in front of them. For an induced-draft plant the arrangement shown in Fig. 39 is well adapted. The curve of the fan scroll is continued within the brick foundation and the air or gas is discharged horizontally at the top, whence it may be readily conducted to a chimney. The engine is of the single upright variety already illustrated in connection with the full-housing fans. It carries the fan wheel upon its extended shaft, and is rigidly supported on a substantial brick foundation bonded into the fan foundation. The absence of a bearing in the inlet leaves it entirely unobstructed for the passage of air or gases, the condition desirable for induced draft adoption.

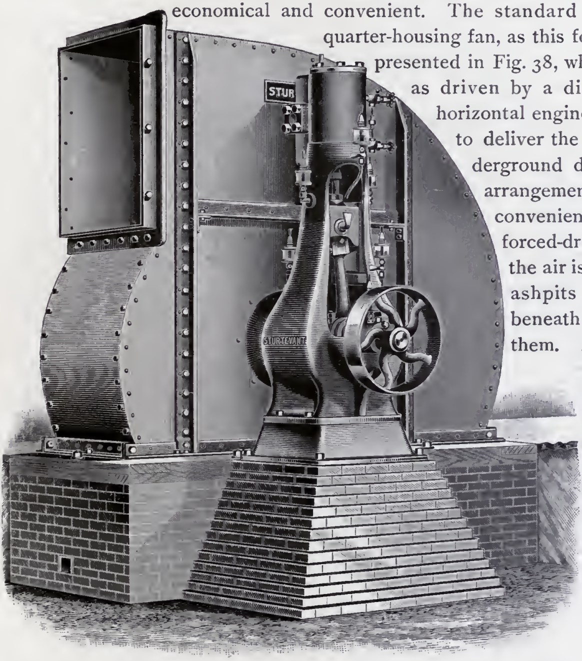

Fig. 39. Steel-Plate Steam Fan with Three-Quarter Housing and Single Upright Engine.

Steel-Plate Steam Fan with Three-Quarter Housing and Double Upright Engine.— One of the most important features desirable in a steam fan applied for induced draft is an engine capable of sustained operation at high speed. The high speed is necessary because of the greater tip velocity which a fan must have in order to produce a given pressure or vacuum when it handles hot gases, while the necessity of continuous operation is evident from the fact that an entire establishment may be dependent for its power upon the draft thus produced. Both of these features are provided in the type of fan illustrated in Fig. 40. The engine is of the double variety, which has been already illustrated in Fig. 32, and there fully described. The large diameter of its cylinders, their short stroke and the cranks set at 180° are all conducive to the development of the maximum power for a given space, and the maintenance of a high relative speed. The wide spacing of the journals on the engine insures the stable overhanging of the wheel, even if of considerable width. Although here shown discharging into an underground duct, a fan of this type can evidently be as readily arranged to discharge in any given direction. When arranged to discharge the air or gas directly upward, it becomes, with its unobstructed inlet, especially adapted for induced draft; for the outlet may be prolonged into a short stack which will be supported by the fan. Any of these fans can be fitted with pulleys instead of engines, and arranged to be driven by belt from any desirable source of power.

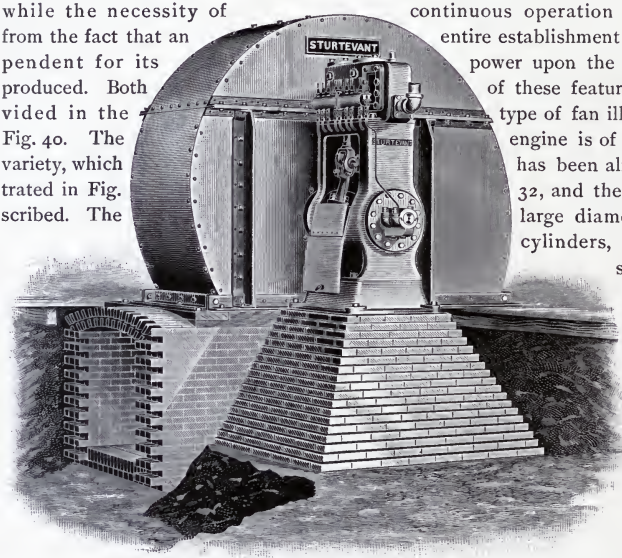

Fig. 40. Steel-Plate Steam Fan with Three-Quarter Housing and Double Upright Engine.

Steel-Plate Steam Fan, Three-Quarter-Housing Type, with Steel-Plate Bottom. — The condition frequently presents itself in induced draft practice where the fan is to be set above the ground floor, and yet it is desired that it be of the three-quarter-housing type. It then becomes necessary to construct the bottom of steel plate in place of masonry. Such an arrangement is shown in Fig. 41. As shown, the I beams of the regular floor construction are so disposed as to furnish the proper support for the fan and its engine, while permitting of the downward projection of the lower part of the housing. By a variety in the heights of the beds for different-sized engines it is possible to locate the centre of such a fan at almost any desired distance above the floor. A complete duplex induced draft plant, with fans of this general type, is presented in Fig. 42. Here the engines are horizontal and the shafts extend through the fan wheels to a central chamber which is kept open to the atmosphere and where they are supported in water-cooled boxes. The fan outlets are angular so as to permit of more ready connection to the stack above. The space between the fans and towards the observer forms a chamber to which the gases are conducted from the boiler. Here the adjustable damper serves to admit them to either fan at will, while the dampers in the connections above operate to open or close the corresponding outlets. As each fan is capable of producing all the draft required for the entire plant, either may be stopped when desired, the dampers so adjusted that no gases will pass through, and the interior thus rendered accessible. The stack is supported on the special steel-beam construction which is practically independent of the fans.

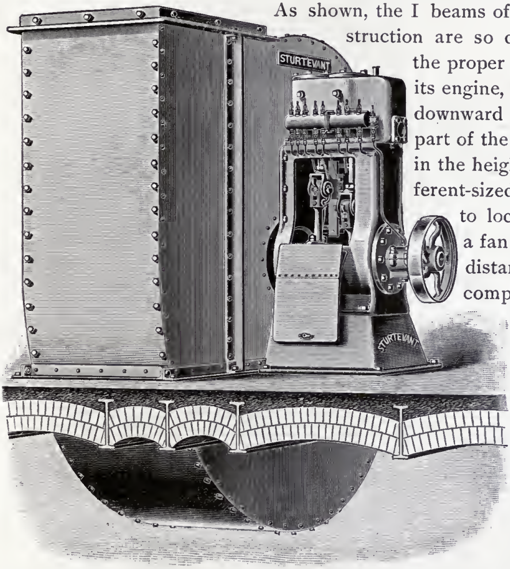

Fig. 41. Steel-Plate Steam Fan, Three-Quarter-Housing Type, with Steel-Plate Bottom.

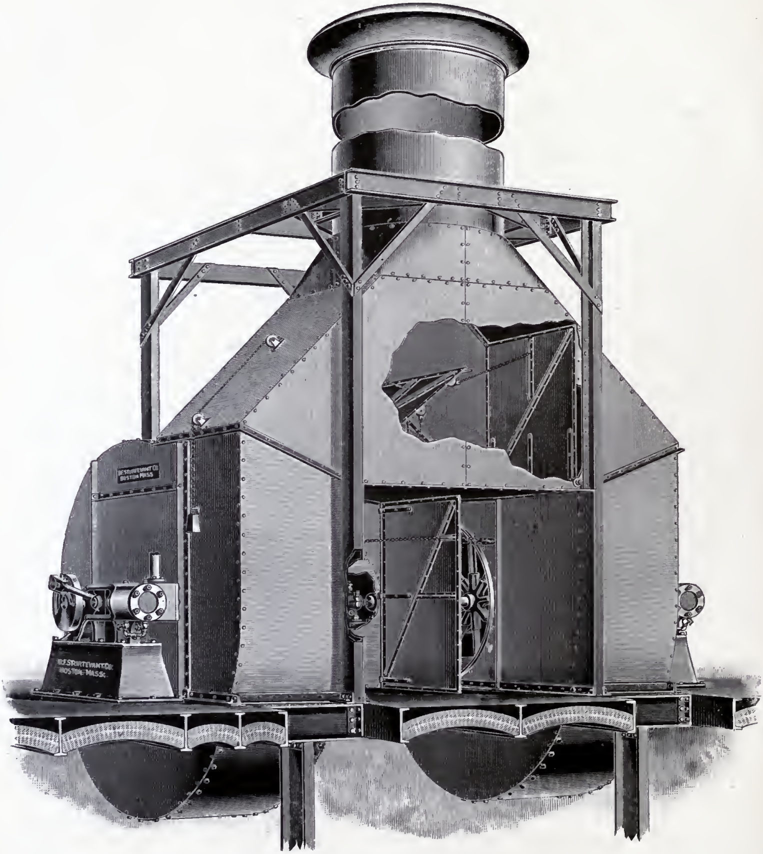
B.F. STURTEVANT CO.
BOSTON MASS.
B.F. STURTEVANT CO.
BOSTON MASS.

FIG. 42. SPECIAL DUPLEX STEEL-PLATE STEAM FAN, THREE-QUARTER-HOUSING TYPE, WITH STEEL-PLATE BOTTOM AND HORIZONTAL ENGINES.

Electric Fan, "Monogram" Pattern. — Conditions may arise in boiler practice in which an electric fan will prove of especial value for draft production. Such a condition frequently exists where the boilers are devoted solely to producing steam of low pressure for heating purposes, but where the draft is insufficient. Manifestly the low pressure makes it practically impossible to introduce a steam fan to assist in the draft production, unless the engine have a cylinder of extraordinary size. An electric fan, however, if installed where a power circuit is constantly available, becomes practically independent of the boiler itself and may be as readily operated when starting the fires as when steam is up. In small plants, with possibly only a single boiler, the purchase of a steam fan, even if the steam pressure is sufficient, is almost out of the question; for if made small enough to do only the work required of it, it becomes almost impracticable. But an electric fan may be furnished at a reasonable cost and of acceptable efficiency, in comparatively small size. Such a fan is that illustrated in Fig. 43. The shell is of cast iron, and of the Monogram type already illustrated in Fig. 16. It furnishes a very rigid structure, to which the motor is attached. This is of the bi-polar type, substantially contained within the heavy wrought-iron circular fieldpiece. The shaft runs in two ring-oiler bearings, and throughout the machine is the product of the most careful design and construction. Its form permits of the fan being placed in any position desired, with the feet uppermost, for instance, or attached to a vertical wall, so that the air may be discharged directly upward or downward, according as the fan may be set. The application of such a fan may save almost the entire cost of an additional boiler when a sufficiently high rate of combustion to accomplish the desired results in the generation of steam is impossible under existing conditions.

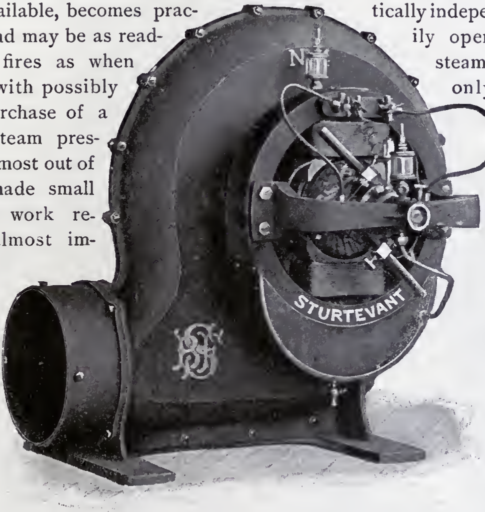

Fig. 43. Electric Fan, "Monogram" Pattern.

Electric Fan, Steel-Plate Pattern. — For comparatively large plants where an electric fan is employed the pattern presented in Fig. 44 is well adapted. The shell is of the same steel-plate construction as the steel-plate exhauster already shown in Fig. 22. The motor shown is of the independent bi-polar type, but in large sizes a multi-polar machine is employed. For small sizes the circular form shown in the preceding illustration is also adaptable. This type lends itself to a great variety of arrangements.

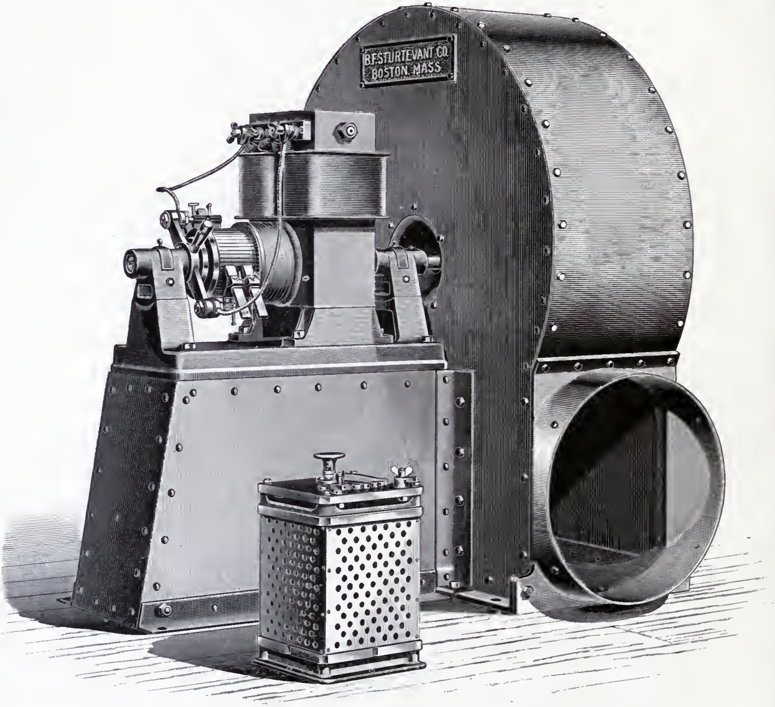
B.F.STURTEVANT CO.
BOSTON, MASS.

Fig. 44. Electric Fan, Steel-Plate Pattern.

CHAPTER XIII.

APPLICATION OF THE STURTEVANT FANS FOR MECHANICAL DRAFT.

The principles of mechanical draft have already been discussed, its advantages shown and various types of Sturtevant fans designed for this specific work have been presented. It now remains to illustrate the manner of application of such fans, and to indicate the economical results obtained by their use. It has already been stated that this Company advocates no particular system or method of application to the exclusion of others, but that it is interested in all arrangements of which a fan forms a part.

In the most impartial manner possible there are here presented descriptions and reports of tests of various systems, together with illustrations of the plants themselves and details of the application. Where a fan alone is concerned this Company is alone responsible (provided that the fan is properly operated), and assumes full credit for the results. But where the fan, of Sturtevant manufacture, is used in connection with air or water heaters, stokers, special grates, or the like, introduced by others as a part of their system, responsibility and credit are assumed only in so far as they directly relate to the fan, although the system may in reality be entirely dependent upon the use of the fan for its successful operation. Therefore, the statements regarding such systems are, so far as possible, quoted from reliable sources and the references given. This is done in the earnest endeavor to present each independently upon its merits, and thereby avoid all appearance of a tendency to draw comparisons.

In the selection of these various plants for presentation the object has been, first, to present only those of which the Sturtevant fan forms a part, and, second, to make them illustrative of as many different arrangements as possible. It has been extremely difficult to choose between the large number of plants which have been installed, and therefore those presented must be considered as only typical of the classes which they represent. They all, however, serve to show in their various ways the manifest advantages of mechanical draft, and to at least suggest the arrangements which are possible under the different conditions which present themselves in practice. They are likewise evidence of the ability of this Company to furnish fans and engines in many types and sizes, each best adapted to the particular requirements.

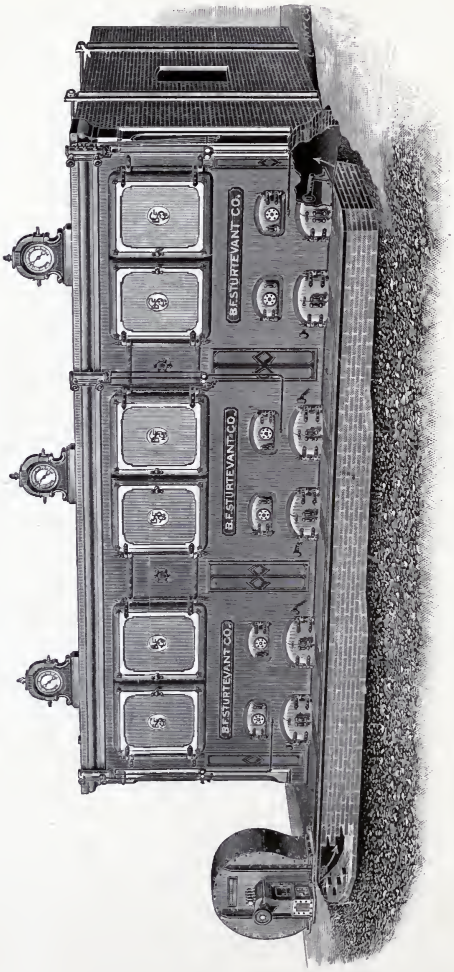
B.F. STURTEVANT CO.
B.F. STURTEVANT CO.
B.F. STURTEVANT CO.

FIG. 45. TYPICAL ARRANGEMENT OF THE STURTEVANT STEAM FAN FOR THE PRODUCTION OF UNDER-GRATE FORCED DRAFT.

Typical Arrangement of the Sturtevant Steam Fan for the Production of Under-Grate Forced Draft.— In Fig. 45 is presented one of the simplest arrangements of a mechanical draft plant. The fan, which is of the steel-plate pattern with angular downward discharge, is driven by a direct-connected enclosed double upright engine. The air is discharged through the entire open bottom into the underground brick duct extending along the front of the battery of boilers. From this duct smaller branches, two to each boiler, extend to the ashpits, to which the air is admitted in the requisite amount through ashpit dampers of the type shown in Fig. 46. There is thus maintained within the ducts and ashpits a pressure greater than that of the atmosphere by an amount dependent upon the speed of the fan, which may be regulated at will. If convenient in such an arrangement, the main duct can be carried immediately beneath the boilers or at any desirable distance in front of them. The fan may be as readily placed in any other position than that shown, and, in case floor space is not available, may be located on top of the boilers and arranged to discharge directly downward into the underground duct. In fact, the possible arrangements are almost without number, for the fan may be of any type, driven by direct-connected engine or electric motor, or by belt from an independent engine, or from a line shaft, and may be located in any convenient position.

Ashpit Dampers. — The success of an under-grate system of forced draft depends largely upon the method adopted for admitting the air to and distributing it within the ashpit. Under ordinary conditions it will not do to admit it through the bottom and allow it to be forced directly upward against the grates. It must first be so deflected as to move upward in substantially equal volume at all parts of the grate. To accomplish this result the form of damper shown in Fig. 46 is employed. As here shown, it is placed in the sloping front of the ashpit bot-

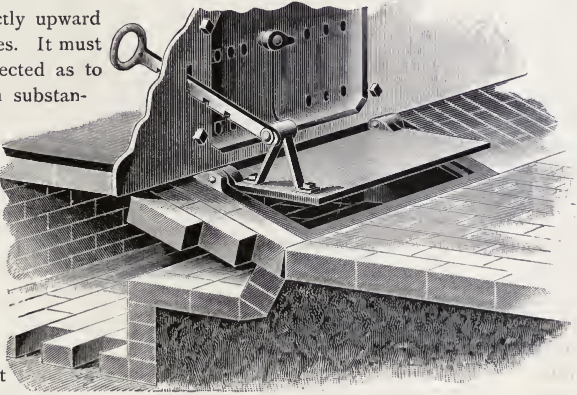

FIG. 46. STURTEVANT ASHPIT DAMPER IN BOTTOM OF ASHPIT.

tom and consists of a frame thoroughly imbedded in the brickwork, a door hinged thereto and a handle bar for operating it. The door is countersunk in the frame so that when closed the top is perfectly flush and the ashes may be readily raked over it. The handle bar is notched so as to permit of adjustment to suit the conditions. In the form here shown the arrangement is readily applicable to an old boiler, for the ducts may be easily constructed without affecting the boiler setting, and the handle bar may be introduced through a single opening which may be readily made in the boiler front.

When the mechanical draft plant forms a part of the original installation, that is, when the use of mechanical draft is decided upon before the boilers are erected, the type and arrangement of ashpit damper shown in Fig. 47 is exceedingly convenient. The air duct is in this case constructed within the bridge wall, and the dampers (one or more to each boiler) are placed as shown. Each damper is provided with an extended shaft and lever so that it may be operated from the boiler front by means of the long handle bar. This is placed at the side, so as not to interfere with the doors. As the air leaves the duct and passes through the damper it is turned downward. It spreads over the entire bottom of the ashpit, thence rising in an even volume at low velocity to the grate above. All possibility of local air blast upon the grate is thus avoided, and the combustion is greatly accelerated, but maintained perfectly regular.

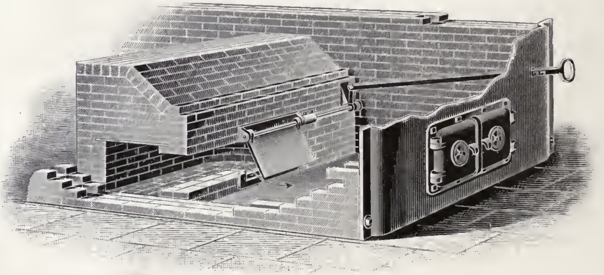

Fig. 47. Sturtevant Ashpit Damper in Bridge Wall.

The importance of proper arrangements for admission and distribution of the air cannot be too greatly emphasized, for upon them practically depends the success of the system. By the introduction of reasonably large ducts and branches the velocity can be kept sufficiently low to prevent its direct impingement upon any portion of the grate when it escapes from the provided opening into the ashpit. The adjustable feature of the damper, together with any device which may be provided for proportioning the speed of the fan to the draft requirements, makes it possible to absolutely control the air supply and draft pressure.

Crystal Water Company, Buffalo, N. Y.— The arrangement illustrated in Fig. 48 serves to emphasize the simplicity of a forced-draft application to a small boiler plant, and is an excellent illustration of an under-grate system. It consists of two 100 horse-power horizontal return tubular boilers supplied by a regular Sturtevant 3 x 4½ steel-plate steam fan. The air from the fan is discharged into a duct extending immediately behind and formed in part by the bridge walls of both boilers, while the air is admitted to the ashpits through special dampers operated by levers, substantially as shown in Fig. 47. One of the levers, extending forward to the front of the boiler, is shown in the cut. The speed of the engine is controlled by a series of regulating and reducing valves introduced under the Beckman system.

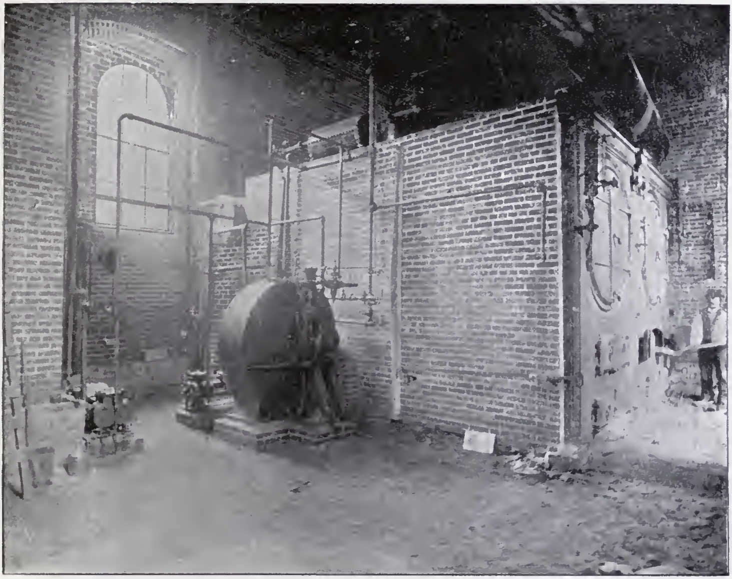

FIG. 48. FORCED-DRAFT PLANT AT CRYSTAL WATER COMPANY, BUFFALO, N. Y.

It is stated[1] that "by this system we are able to hold our steam pressure

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[1] Crystal Water Company, Buffalo, N. Y. Letter of March 12, 1896, to B. F. Sturtevant Co.

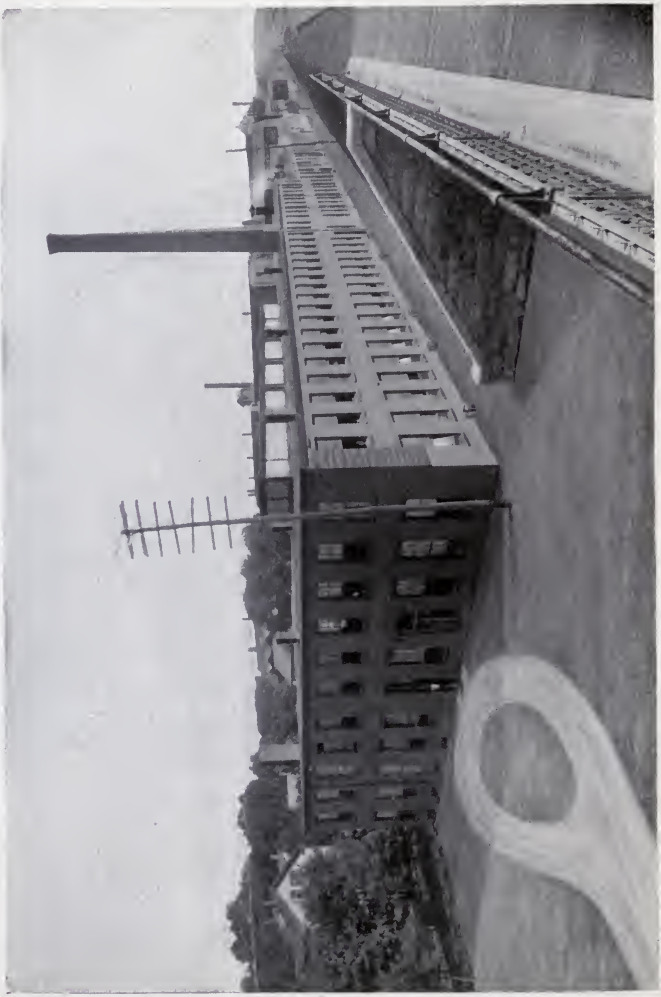

FIG. 49. VIEW OF B. F. STURTEVANT CO.'S WORKS, JAMAICA PLAIN, MASS., SHOWING RELATIVE SIZES OF CHIMNEY AND INDUCED-DRAFT PLANT STACK.

within four or five pounds of a given point and a large saving in steam is effected, as the blower runs at low speed most of the time. We use a four-to-one mixture of hard-coal screenings and soft run of mine, costing us about 30 cents less per ton than clear soft coal, while entirely obviating the smoke customary where soft coal is used clear. Our boiler capacity is largely increased, and our labor of firing is considerably lessened.

"Our draft pressure is from ½ to 1½ inches of water, according to the requirements. We consume from 20 to 30 pounds of coal per square foot of grate surface per hour when running under full load at 80 pounds steam pressure and a speed of from 400 to 480 revolutions per minute. We have a large stack with good natural draft, but can neither burn the coal as efficiently nor as fast as necessary when forced draft is not in use."

As indicative of the efficiency of mechanical draft to prevent smoke, the following statement[1] is presented: "The city has recently passed a smoke ordinance compelling all concerns to put on smoke consumers. The committee, after examining our plant a few days since, decided that it was not necessary for us to put on a smoke consumer on account of the forced draft used."

B. F. Sturtevant Co., Jamaica Plain, Mass.— This plant presents an excellent exemplification of a number of the advantages of mechanical draft. Radical changes in the arrangement of the works necessitated the removal of the boiler plant to a position over 100 feet from its original location and that of the chimney which had previously served to produce the draft. The distance was too great to permit of the further use of the chimney, and its removal was out of the question. It was therefore permitted to stand as a useless monument to a discarded method of draft production. A new chimney in the desired location would have cost about $1,450, and would have occupied valuable floor area. It was, therefore, decided to introduce a fan driven by direct-connected engine and designed to operate on the induced system. Its market value, complete with the stack, was about $700, which may be compared with that of the chimney, manifestly to the disadvantage of the latter. The fan was placed above the boilers and discharged through a small short stack extending just above the roof. It was provided with a by-pass for emergencies. The relative proportions of the stack and chimney are clearly shown in Fig. 49, which presents a perspective view of the front side of the works. The stack appears just beyond and to the right of the chimney. Comment is unnecessary.

The boiler plant, which is of about 260 horse-power nominal rating, consists of three boilers arranged as shown in Fig. 50. Two of these boilers are 66

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[1] Crystal Water Company, Buffalo, N. Y. Letter of July 30, 1897, to B. F. Sturtevant Co.

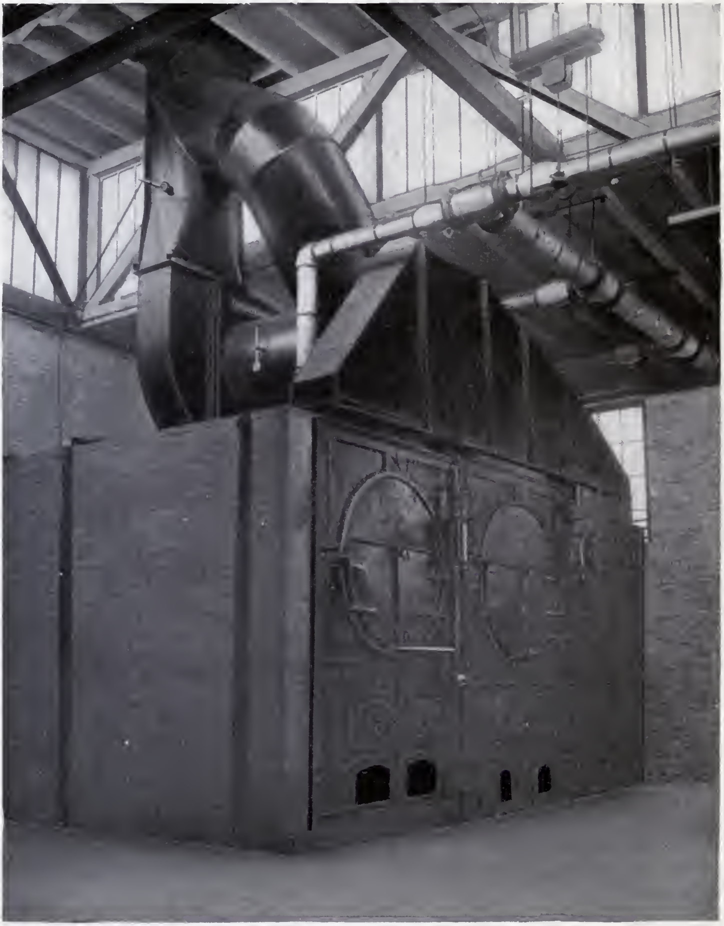

FIG. 50. INDUCED-DRAFT PLANT AT B. F. STURTEVANT CO.'S, JAMAICA PLAIN, MASS.

inches in diameter, each containing 123 tubes 3 inches in diameter by 15 feet long. The third boiler is 60 inches in diameter, and contains 64 tubes 3½ inches in diameter by 15 feet long.

The fan is a special 90-inch Sturtevant exhauster, to which is attached, upon the farther side, and not visible in the illustration, a 5 x 4 double upright Sturtevant engine. The speed of the engine, and consequently the intensity of the draft, is regulated by a special device which serves to open the throttle and increase the steam supply in proportion as the steam pressure is reduced. It is so adjusted that a decrease of about one pound in the steam pressure serves to change the speed from minimum to maximum, or *vice versa*. The corresponding steam and draft pressures under working conditions, as shown by continuous graphic records from special instruments, are presented later in Figs. 51 and 52.

Shortly after the installation of this plant, in which the surface ratio was not changed from that previously existing with chimney draft, comparative tests were made under the direction of Prof. Peter Schwamb, of the Massachusetts Institute of Technology.

Three separate tests were conducted upon successive days, under the following conditions: —

Test No. 1. Run on middle boiler, dampers to other two boilers closed and packed, fires drawn, and all steam and water connections broken. Fan engine under control of special regulator. Coal as indicated in Table No. 121.

Test No. 2. Conditions same as in No. 1, except that fan engine was fitted with Water's governor, and damper in inlet connection to fan was operated by a Locke damper regulator. Coal as indicated in Table No. 121.

Test No. 3. Run on three boilers. Fan engine regulated as in No. 1. Fire even, not drawn, but running start and stop were made. Coal as indicated in Table No. 121.

The air supply was determined by anemometer readings at ashpit doors.

Table No. 121. — Analysis of Coal, Coal and Ash (on Grate) and Ash — B. F. Sturtevant Co. Test.

Number of Test.	COAL ANALYSIS.						Coal and Ash on Grate at Close of Test.	Ash.
	C	H	Moisture.	Ash or Earthy Matter.	O	N	C	C
1	81.9	5.18	0.5	7.61	3.3	1.5	21.8	30.52
2	81.6	5.0	1.04	7.47	3.2	1.7	21.3	35.65
3	76.3	2.0	5.4	12.3	——	——	——	——

The coal used in Tests Nos. 1 and 2 was clear George's Creek Cumberland, costing $3.65 per ton of 2,000 pounds. In Test No. 3 this was mixed half-and-half with yard screenings costing $2.00 per ton, making the cost of the mixture $2.82½ per ton.

Table 122. — Results of Tests of Induced-Draft Plant at B. F. Sturtevant Co., Jamaica Plain, Mass.

		Test No. 1.	Test No. 2.	Test No. 3.
Duration	hours,	11.5	11.52	11.55
Average steam-gauge pressure	pounds,	79.4	79.3	78.9
Average temperature of feed water	degrees,	175.3	189.5	148.6
Grate surface	square feet,	30.25	30.25	85.5
Water-heating surface	square feet,	1,473.4	1,473.4	3,884.6
Nominal rating, at 15 square feet of heating surface per horse-power,	horse-power,	98.2	98.2	258.9
Ratio of water-heating surface to grate surface		48.7	48.7	45.4
Coal burned, including coal equivalent of wood	pounds,	7,461.7	5,723.3	13,613.5
Total combustible consumed	pounds,	7,005.8	5,263.3	12,691.1
Air per pound of coal burned	pounds,	20.12	20.80	21.81
Air per pound of combustible	pounds,	21.43	22.62	23.39
Quality of steam, saturated steam taken as unity		0.996	0.995	0.995
Water actually evaporated, corrected for quality of steam,	pounds,	70,117.4	54,862.	114,568.
Equivalent water evaporated into dry steam from and at 212°,	pounds,	75,326.9	58,115.	126,313.
Equivalent water evaporated into dry steam from and at 212°, per pound of coal burned,	pounds,	10.10	10.15	9.28
Equivalent water evaporated into dry steam from and at 212° per pound of combustible,	pounds,	10.75	11.04	9.95
Actual horse-power at 34.5 pounds per hour, from and at 212° per horse-power,	horse-power,	189.6	146.3	317.0
Coal burned per square foot of grate per hour	pounds,	21.45	16.45	15.09
Water evaporated from and at 212° per square foot of heating surface per hour,	pounds,	4.45	3.43	3.08
Average draft pressure in ashpit	inches of water,	0.144	0.078	0.057
Average draft pressure at back	inches of water,	0.640	0.372	0.581
Average draft pressure at fan	inches of water,	1.253	1.743	1.342
Temperature of boiler room	degrees,	80.9	77.7	80.4
Temperature at uptake	degrees,	473.9	436.3	468.4
Temperature at fan	degrees,	389.5	332.5	438.7
Revolutions of fan per minute		291.6	407	352.3
Steam consumed by fan engine per hour	pounds,	272.3	425.3	438.2
Steam used by fan engine per pound of water evaporated (dry steam, 212°)	pounds,	0.0416	0.0769	0.0366

The general conditions and results of the tests are given in Table No. 122. The purpose of producing all the steam by means of one boiler in Tests Nos. 1 and 2 was to give an opportunity to force the boiler well above its rated capacity. As a result, in Test No. 1 it exceeded its rating by 93 per cent, and in Test No. 2 by 49 per cent. These values are to be taken into account in considering the economic results. The coal burned per square foot of grate was in the first test brought up to 21.45 pounds; but, owing to the fact that no special arrangement of the grates was adopted, the air supply was 20.12 pounds per pound of coal, somewhat higher than it might otherwise have been. The evaporation from and at 212° per pound of combustible shows a difference of only about 2½ per cent between the two tests, even under the different conditions.

The difference in the steam required to operate the fan engine under the three tests is to be noted. In Test No. 1 the engine was automatically controlled in the usual manner to suit the requirements, but in Test No. 2 the fan, operating at constant speed under the influence of the Water's governor, presented approximately the condition with a chimney which produces a constant maximum intensity of draft which is adapted to the requirements by shutting off a portion of the volume by means of a damper. As a consequence, the steam expenditure for draft is greater in the latter case. The decreased relative amount of steam used to operate the fan with a larger plant is shown by Test No. 3. As in the original design it was determined that the exhaust steam from the fan engine was to be utilized for heating the feed-water, the question of efficiency of the engine was of minor account. In fact, with this in view an engine of simple construction, capable of continuous operation, was adopted, rather than one which would show higher efficiency but not be so serviceable for this work. Hence the somewhat high values for the water per horse-power per hour, shown in Table No. 123, are not surprising. Had the consumption been

Table No. 123.—Results of Tests of Fan Engine at B. F. Sturtevant Co.'s, Jamaica Plain, Mass.

		Designation of Test.			
		A	B	C	D
Duration	minutes, seconds,	11:0	2:15	10:40	8:55
Revolutions per minute		438	449	443	436
Temperature at fan	degrees,	394	387	392	366
Indicated horse-power		8.20	8.26	7.99	8.22
Water per I. H. P. per hour	pounds,	63.5	65.4	66.9	64.4

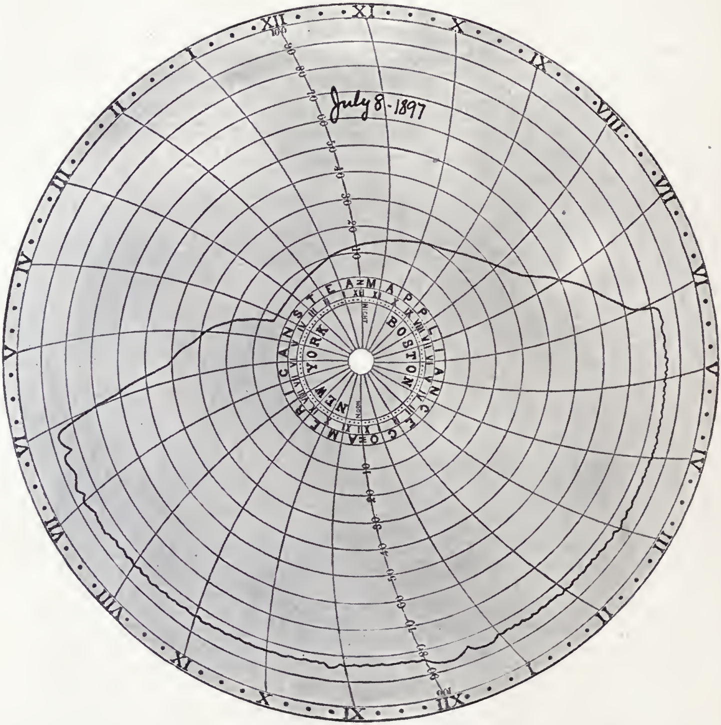

FIG. 51. STEAM-PRESSURE CHART FROM BOILER PLANT AT B. F. STURTEVANT CO.'S, JAMAICA PLAIN, MASS.

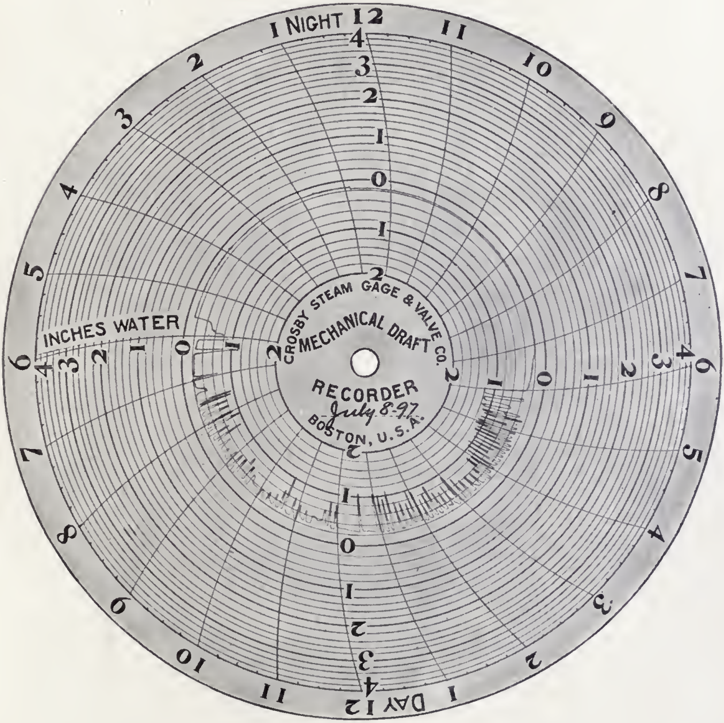

Fig. 52. Draft-Pressure Chart from Induced-Draft Plant at B. F. Sturtevant Co.'s, Jamaica Plain, Mass.

only 30 pounds, the fan engine would have required in Test No. 3 only 1.7 per cent of the total steam produced by the boiler plant. The results of tests of other and larger plants, presented in this chapter, show much more economic relations.

The efficiency of the combustion is manifestly one of the most important matters in connection with a boiler plant. In the case of this test, flue-gas samples were continuously taken from a point at the back end of the boilers where the gases entered the tubes, and also in the uptake just after they had left the tubes. These samples were analyzed by means of the Orsat apparatus, and the results obtained as presented in Table 124. The amount of carbonic oxide (CO) was at all times too small to be detected by the apparatus. Numerous observations of the top of the chimney stack also failed to show the presence of smoke, except for a few seconds after each firing. The photograph from which Fig. 49 was reproduced was taken during working hours, and serves to show the absence of smoke.

Table No. 124. — Analysis of Continuous Gas Samples — B. F. Sturtevant Co. Test.

Number of Test.	Duration.	BACK END.				UPTAKE.			
	Hours.	CO_2	O	CO	N +	CO_2	O	CO	N +
1	9	6.72	10.72	0	81.33	5.81	13.53	0	80.68
2	10	7.50	11.09	0	81.31	5.80	13.88	0	80.32
3	9.5	6.71	12.78	0	81.57	5.75	14.14	0	80.11

In connection with this plant there is in operation a continuous steam pressure recorder and a mechanical draft recorder of the form previously illustrated in Fig. 9. This latter is connected to the inlet pipe which leads to the fan, and, therefore, records the total pressure rather than the difference between over and under-grate pressures. Two typical and corresponding charts from these instruments are reproduced in Figs. 51 and 52. The steam pressure is maintained almost absolutely constant, while the draft pressure necessary to secure this result varies suddenly and between considerable limits. The instant response of the fan and the range through which it operates are evidently essential to the results obtained. The fluctuations of the fan speed coincide closely with the periods of firing, and thus serve to furnish a large volume of air under greater intensity at just the instant when most required because of the increased discharge of inflammable gases.

Table No. 125.—Results of Full-Power Forced-Draft Contract Trials of Vessels of U. S. Navy Equipped with Sturtevant Fans.

CLASS AND NAME.	Dimensions.	Displacement, in Tons.	Mean I. H. P. of all Machinery.	Speed per Hour, in Knots.	Number of Sturtevant Blowers on Vessel.	Total H. P. Required to Drive Blowers.	Air Pressure in Fire Room, in Inches of Water.	Coal per Hour per Square Foot of Grate, in Pounds.
Cruiser Detroit,	L. 257 ft. 0 in. B. 37 ft. 0 in. D. 14 ft. 5¾ in.	2,068	5,227	18.71	6	18.9	0.8	
Cruiser Montgomery,	L 257 ft. 0 in. B. 37 ft. 0 in. D. 14 ft. 0¼ in.	2,091	5,527	19.06	6		1.4	
Gunboat Machias,	L. 190 ft. 0 in. B. 32 ft. 0 in. D. 12 ft. 0¾ in.	1,068	1,873	15.46	2	23.2	0.47	38.08
Gunboat Castine,	L. 190 ft. 0 in. B. 32 ft. 0 in. D. 12 ft. 0¾ in.	1,068	2,199	16.03	2	16.9		42.6
Monitor Monadnock,	L. 259 ft. 6 in. B. 55 ft. 10 in. D. 14 ft. 6 in.	3,990	3,000	14.5	2			
Cruiser New York,	L. 380 ft. 0 in. B. 64 ft. 3 in. D. 23 ft. 10¾ in.	8,480	17,401	21.0	14	90.2	1.6	
Cruiser Brooklyn,	L. 400 ft. 6 in. B. 64 ft. 0 in. D. 21 ft. 10½ in.	8,150	18,770	21.9	14	269.1	2.26	
Cruiser Columbia,	L. 411 ft. 7¼ in. B. 58 ft. 2¼ in. D. 22 ft. 5 in.	7,350	18,509	22.8	18	166.3	0.73	
Battleship Massachusetts,	L. 348 ft. 0 in. B. 69 ft. 0 in. D. 24 ft. 6 in.	10,265	10,403	16.21	10	107.2	0.99	
Battleship Indiana,	L. 348 ft. 0 in. B. 69 ft. 3 in. D. 24 ft. 0 in.	10,225	9,738	15.55	10	52.8	0.96	
Battleship Iowa,	L. 360 ft. 0 in. B. 72 ft. 2¾ in. D. 24 ft. 0½ in.	11,363	12,105	17.09	8	104.6	0.99	

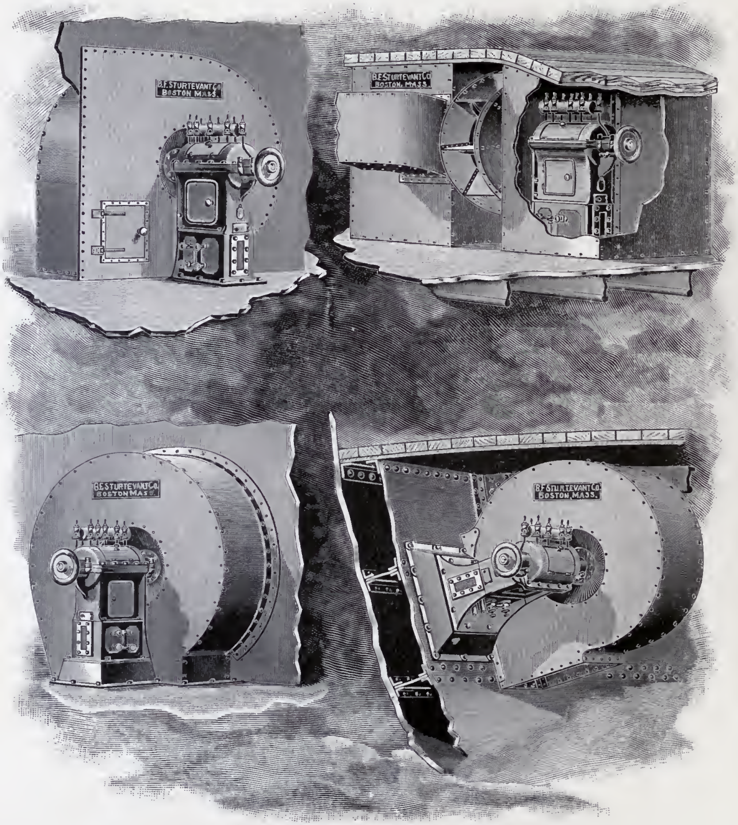
B.F. STURTEVANT Co. BOSTON MASS.
B.F. STURTEVANT Co. BOSTON, MASS.
B.F. STURTEVANT Co. BOSTON MASS.
B.F. STURTEVANT Co. BOSTON, MASS.

FIG. 53. TYPES OF STURTEVANT SPECIAL STEAM FANS APPLIED FOR FORCED DRAFT ON VESSELS OF U. S. NAVY.

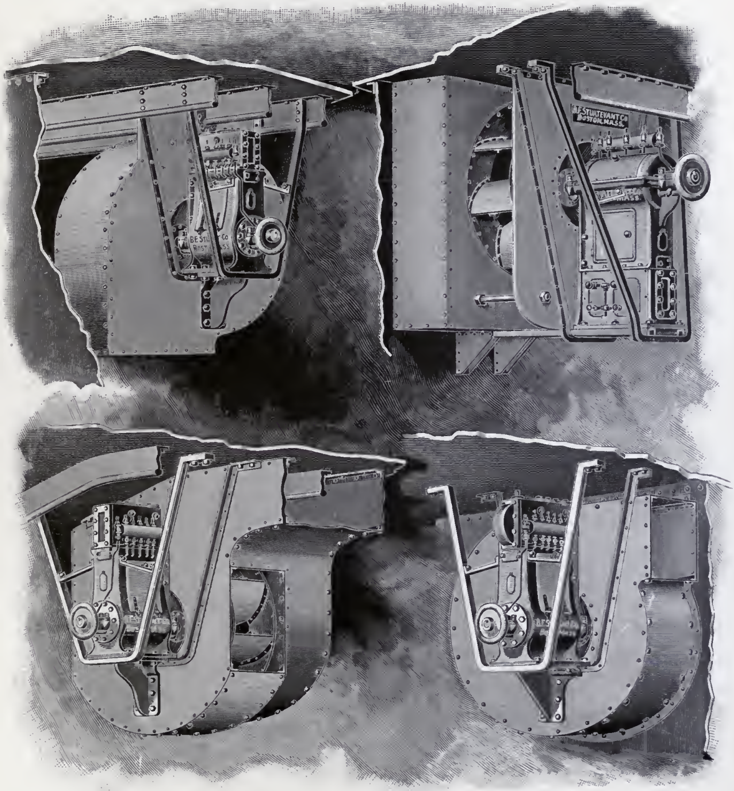
B.F. STURTEVANT Co
BOSTON, MASS.

FIG. 54. TYPES OF STURTEVANT SPECIAL STEAM FANS APPLIED FOR FORCED DRAFT ON VESSELS OF U. S. NAVY.

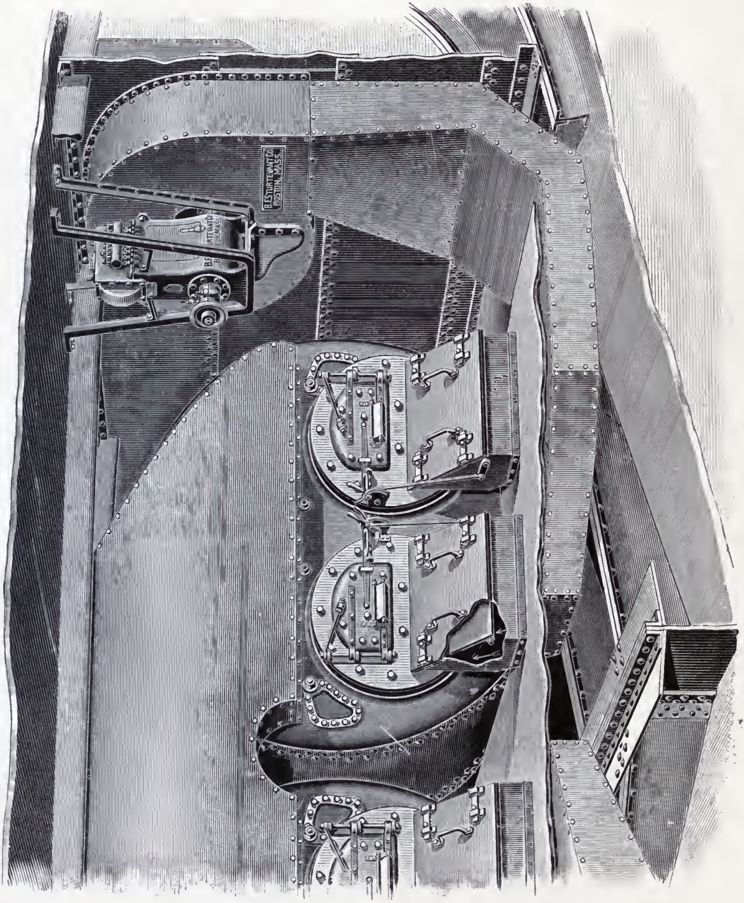
B.F. STURTEVANT CO.
BOSTON, MASS.

Fig. 55. Application of Sturtevant Fan in Conjunction with Kafer's Closed Ashpit System of Forced Draft.

U. S. Navy.—The importance of forced draft as one of the elements of success in the modern warship has already been pointed out. By its means it is possible to instantly and greatly increase the capacity of the boilers, to thereby make greater speed possible, and to entirely change the aspect of an engagement. This increased boiler capacity is secured by the introduction of a few light and comparatively inexpensive fans instead of greatly enlarging the boilers themselves, for the purpose of providing reserve capacity, as would otherwise be necessary. The Navy Department of the United States was among the first to recognize the importance of such an arrangement, and early called upon this Company to design and furnish special fans for draft production. It being the intention to use these fans principally as a reserve, it was seen to be desirable that they should be capable of operation at high speed, both to produce the required air pressure (of frequently 4 inches of water at the fan) and to deliver the largest possible amount of air while occupying the minimum of space. High speed was obviously imperative. This was secured by designing the special type of double-cylindered upright enclosed engine already shown in Fig. 32.

The types of Sturtevant fans presented in Figs. 53 and 54 are so chosen because they are representative, not because they begin to cover all the forms designed for this special work. The results of full-power forced-draft contract trials on the vessels for which these particular fans were furnished are presented in Table No. 125, which covers only a few of the vessels of the U. S. Navy equipped with Sturtevant fans. In fact, nearly 400 Sturtevant fans have been installed for various uses on over 50 vessels of the U. S. Navy. A comparison of figures will show that on an average the horse-power required to drive the fans is only about 0.88 per cent of the mean indicated horse-power of all the machinery, omitting the case of Cruiser Brooklyn, where it is excessively high. The average is only about 0.75 per cent.

U. S. S. Swatara.—This vessel is equipped with Sturtevant fans for the production of draft upon the closed ashpit system. The fans are two in number, each having a wheel 50 inches in diameter by 24 inches wide, enclosed in a steel-plate case arranged so that the air may be discharged directly downward. Each fan is driven by a direct-connected 5 x 5 double enclosed upright engine. The results of three tests of these fans have been given in Table No. 114.

The fans were installed in accordance with the designs of Mr. John C. Kafer, a general illustration of whose system, as applied to a marine boiler and supplied with a Sturtevant fan, is presented in Fig. 55.

The ashpits are closed by means of light iron doors, held in place by buttons at the corners and made air tight by asbestos gaskets. The air from the blower passes through the duct to these closed ashpits, whence it has two means of escape,— either through the fuel on the grates, or through openings in the dead plates to the space between the inner and outer plates of the fire doors.

The operation of the system may be thus further described[1]: "As the air pressure in the space between the inner and outer plates of the furnace door is from one-fourth to one-half inch of water greater than the pressure of gases in the furnace, no gases can escape from the furnace, as the furnace door is air packed. The air pressure between the plates of furnace door being greater than the air in the fire room or the gases in the furnace, all leakage is from this space into the furnace and to the fire room, and is fresh air, making a better combustion of gases in the furnace, at the same time keeping the furnace door and front quite cool.

"A hinged valve in the air duct, which is shown in the illustration as closed, shuts off or regulates the supply of air to each furnace, and is operated by a lever from the fire room. When the valve is open, the lever is fitted to lock the closed furnace door.

"This device was first fitted to the U. S. S. Alliance in 1886, and has been in constant use while cruising under steam, and is satisfactory in every particular. In this ship the cost of two boilers is saved,— using four instead of six,— decreasing the weight and space occupied by the boilers, enabling the ship to carry about 40 tons more coal, giving her increased endurance, at the same time increasing her maximum power over 50 per cent above the power with six boilers under natural draft. Since the U. S. Steamers Swatara and Kearsarge have been similarly fitted [in both cases with special Sturtevant fans driven by direct-connected upright engines], and are now cruising with two less boilers, increased coal capacity and 50 per cent more horse-power.

"In a test of the boilers of the U. S. S. Swatara by a board of naval engineers, an average air pressure of 5 inches of water in the air duct, with an average consumption of 44.6 pounds of anthracite coal per square foot of grate per hour, was maintained for 6 hours; the maximum rate of combustion was 52 pounds per square foot of grate. The evaporative efficiency of the boiler under forced draft during a 12-hour test was 9.68 pounds of water per pound of combustible. The boilers of the Swatara are cylindrical, with return fire tubes over the furnaces 9 feet in diameter, 9 feet long, with two furnaces in each boiler 34 inches diameter, having grates 5½ feet long; the proportion of heating to grate surface is 23 to 1,— a very low proportion for forced draft, the boilers having been designed for natural draft only."

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[1] Kafer System of Forced Draft. The Marine Journal, New York, March 2, 1889.

American Line Pier 14, N. R. International Navigation Company, New York, N. Y.— This plant serves as an excellent illustration of a system of induced mechanical draft in connection with a special device for abstracting the heat from the escaping gases and imparting it to the air supplied to the boiler furnaces. The particular system here employed is known as the "Ellis & Eaves," is equipped with a Sturtevant fan, and is thus described[1] : —

"This is a combination of four old principles: 1. Induced (or suction) draft; 2. 'Serve' tubes; 3. Retarders in tubes; 4. The air heated by the waste gases.

"The object is to burn with *safety* and *economy* a much larger quantity of coal than has hitherto been done in a boiler of a given size; consequently to produce the power required with less boiler space, weight, and even first cost, without the troubles which frequently have accompanied high 'forced' draft. In the 'Ellis & Eaves' combination, fans are used by preference, to exhaust the gases from the boilers. This is no new idea, but it has only now been rendered really practicable for high rates of combustion by absorbing the heat of the gases before they reach the fans. This prevents trouble of any kind in the fans. The fan power can be increased to almost any extent, and draft equal to that of a chimney 200, 300 or more feet can readily be thus obtained. . . . In the fans and fan engines used daily for nearly twelve months past with Nos. 7 and 8 marine boilers at the Atlas Works, no trouble whatever has occurred, although for at least half the time the combustion has been over 40 pounds per square foot of grate 5 feet 8 inches long, exclusive of dead plate. Spurts have been made to 60 pounds per square foot of same size grate without trouble, because the gases have not exceeded 450° Fahr. when entering the fans. Nos. 9 and 10 boilers, now being completed, are intended to be capable of burning an average of 60 pounds per square foot of grate 5 feet 8 inches long for any length of time, just as Nos. 7 and 8 can burn 45 pounds now.

"No one questions the use of natural draft. This system is natural draft magnified."

"The Serve tubes, in the first place, absorb a larger quantity of the heat than can be done with plain tubes. . . . A retarder placed in the centre of the 'Serve' tube (whether it be a strip of flat iron or steel twisted into a spiral shape, or a thin plain tube with the fire-box end closed) forces the gases out of the centre of the tubes into the spaces between the ribs, and thus in closer contact with the heat-absorbing surface specially provided. . . . 'Serve' tubes

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[1] The "Ellis & Eaves" Patent Combination Induced Draft. Catalogue, March 1, 1893. John Brown & Co., Limited, Sheffield, Eng.

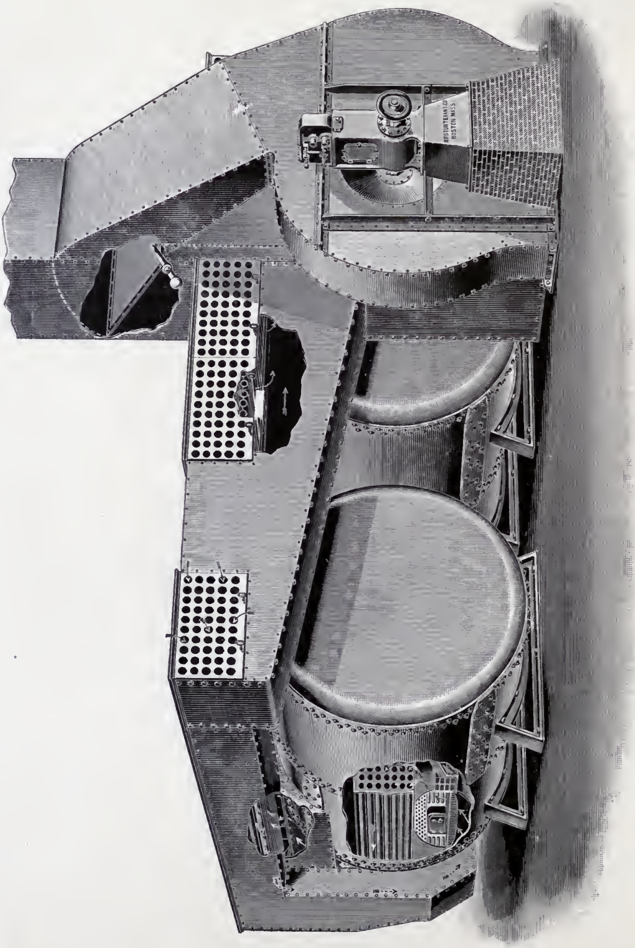
B F STURTEVANT CO
BOSTON MASS

Fig. 56. Induced-Draft Plant on Ellis & Eaves System with Sturtevant Fan, at American Line Pier 14, International Navigation Co., New York, N. Y.

with retarders give at least 15 per cent more steam, or economy, than plain tubes of the same diameter without retarders, and at least 10 per cent more than plain tubes of the same diameter. Whilst with ordinary natural draft retarders cannot be used, they present no difficulty with medium and high artificial draft.

"Heating the air for the furnaces by the heat of the waste gases serves the double purpose of utilizing the latter for producing steam, and of making them harmless when they reach the fan, the heat having been absorbed. Vertical short air-heating tubes have been used in another system for this purpose. In the 'Ellis & Eaves' system horizontal and long tubes are naturally more efficient, and so far it is considered preferable to pass the air through them, instead of exhausting the gases through them.

"Summarized briefly, the benefits of this system are (taking natural draft and plain tubes burning at 15 lbs. per square foot of ordinary size grate as basis), —

"(*a*) An economy of 10 per cent of fuel when burning at 30 pounds on the same size of grate, consequently with half the number of boilers, the space and weight being less than with the natural draft and plain tubes, and the first cost not more.

"(*b*) An equal amount of water evaporated per pound of fuel, when burning at 45 pounds per square foot of the same size grate, consequently with *one-third* the number of boilers; the saving in weight, space and first cost being then considerable. In both cases the amount of steam used by the fans is allowed for in the comparison with natural draft."

As regards its other advantages, it is said[1] that "the *greater cleanliness* of the suction draft is another strong point in its favor, the particles of coal dust and ashes being sucked up through the smokestack instead of being blown all over the ship. The fire room with the Ellis & Eaves draft is much cooler than with either natural, forced or closed stokehold draft, so that the firemen are always in a condition to do better work and more of it without exhaustion. . . . Finally, with the Ellis & Eaves draft there is *perfect combustion and consequently no smoke, no matter what description of coal is used.*"

[1] "When the power plant for the recently constructed pier for the American Line of steamships in New York City was being built, there were installed two single-ended Scotch marine boilers, both of which were fitted with the Ellis & Eaves system; while one boiler was fitted with plain tubes, the other was supplied with Serve patent ribbed steel tubes. [Fig. 56 shows the general manner in which the plant is arranged.] The waste gases are carried

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[1] Marine Journal, New York, June 1, 1895.

from the breeching back over the boilers through two air-heating chambers and finally pass from these through the necessary connections to an 8-foot Sturtevant fan discharging into a 70-foot stack. A by-pass around the fan is provided so that the boilers may operate under natural draft if desired. The air-heating chambers contain a number of 3-inch tubes, each about 12 feet long, through which the air that is supplied the furnaces is drawn. The air is highly heated in its passage by the waste gases from the boiler, and is delivered at a high temperature both above and below the fires, the amount to each being regulated by butterfly valves. As the fan tends to produce a slight vacuum in the furnaces, both the ashpit and the fire doors are made to fit tightly to prevent the in-leakage of cold air. To prevent cold air from entering the furnace the fire door, on being opened, automatically closes a large

Table No. 126. — Results of Tests of Ellis & Eaves Induced-Draft System with Sturtevant Fan, at American Line Pier 14, N. R., New York, N. Y.

ITEMS.		Number and Conditions of Test.		
		No. 1. No. 1 Boiler, Plain Tubes with Retarders.	No. 2. No. 1 Boiler, Plain Tubes with Retarders.	No. 3. No. 2 Boiler, Serve Tubes with Retarders.
Date	1894,	Sept. 28.	Oct. 5.	Sept. 29.
Duration of test	hours,	6	6	6
Average steam-gauge pressure	pounds,	96	95	96
Average temperature of feed water	degrees Fahr.,	157	166	165
Average revolutions of fan per minute,		507	472	468
Total water evaporated	pounds,	72,661	59,708	71,697
Total coal burned	pounds,	11,000	8,920	8,715
Total combustible consumed	pounds,	9,746	7,780	7,717
Coal burned per square foot of grate per hour	pounds,	52.08	42.23	41.26
Water evaporated per pound of coal from and at 212°,	pounds,	7.228	7.274	8.939
Water evaporated per pound of combustible from and at 212°,	pounds,	8.165	8.332	10.097
Average temperature at fan outlet	degrees Fahr.,	625	527	456
Average temperature at fan inlet	degrees Fahr.,	650	547	472
Average temperature at air down-take	degrees Fahr.,	400	347	315
Average vacuum at fan inlet	inches of water,	7.33	6.61	6.45
Average vacuum over fires	inches of water,	3.88	3.46	3.56
Average vacuum under fires	inches of water,	1.72	1.38	1.48
Average vacuum at air down-take	inches of water,	0.62	0.58	0.50

butterfly valve in the uptake, thus shutting off the draft. Retarders, consisting of long strips of thin steel of a width equal to the inside of the tube diameter, and twisted into a helix of about three turns in the length of the tube, are placed in the boiler tubes to retard the flow of gases through them. The Serve tubes also contain retarders.

"Three tests in all were made, all of six hours' duration, the boilers in all instances being fired with No. 1 buckwheat Susquehanna coal. The first test was made with the fan running at 520 revolutions per minute, this causing a vacuum of 7½ to 8 inches on the fan suction. All of the hot air was admitted to the furnaces underneath the grates. The boiler was fired lightly every 5 minutes, and the fires were cleaned slightly about every 45 minutes. They were three-fourths cleaned after 3½ hours' running. Tests Nos. 2 and 3 were made under the same conditions, the fan being run at 360 revolutions at the start and gradually increased so that the maximum speed was attained after 2½ hours' running. In all of the tests the thickness of the fires was estimated at the beginning and at the end of the tests. The water was measured by a previously calibrated Worthington meter. Test No. 2 was made upon boiler No. 1 with plain tubes, and test No. 3 upon boiler No. 2 fitted with Serve tubes." The general results of the three tests are presented in Table No. 126.

Steamer L. C. Waldo.— This vessel, of the Roby Transportation Company, Detroit, Mich., is equipped with the Ellis & Eaves system of induced draft operated by Sturtevant fans. An expert test of the same, conducted by Mr. George C. Shepard, is thus reported[2]: —

"In the table of performances of modern lake steamers in the Blue Book of American Shipping, the tests of fourteen steamers are included. On only four of this number does the coal consumption per horse-power per hour fall below two pounds. On the other ten it ranges from 2.02 pounds to 2.64 pounds, the average fuel consumption being 2.22 pounds of coal per horse-power per hour. It is almost needless to say that none of these ten steamers use artificial draft.

"The test of the L. C. Waldo, published herewith, will be of interest to vessel owners and engineers, as it shows a consumption of 1.88 pounds of coal per horse-power per hour, including all tests,— but excluding one test, a short one made with the dampers closed, as an experiment. The result is 1.79 pounds, or a gain of some 20 per cent, as compared with the average of the ten modern lake steamers above referred to.

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[1] The Engineering Record, New York, Jan. 5, 1895.

[2] Marine Review, Cleveland, O., Oct. 22, 1896.

"The inventors of the Ellis & Eaves system will no doubt call attention to the fact that the Waldo's boilers are equipped with plain tubes and not with Serve's ribbed tubes, which they claim is the complement of their draft system, and that by their use an additional 10 or 15 per cent could be obtained. Allowing 10 per cent for the Serve's tubes, the Waldo's consumption would be 1.61 pounds; and, allowing for auxiliaries, it is not unreasonable to claim a consumption of only 1½ pounds for this system.

"The aggregate heating surface of the two boilers is 6,230 square feet and the aggregate grate surface is 120 square feet, making the ratio 51.92. The peculiar feature in this steam plant is to be found in the Ellis & Eaves system of induced draft. In this system the products of combustion are drawn from the boilers by fans.

Table No. 127.—Results of Tests of Steam Plant on Steamship L. C. Waldo with Ellis & Eaves Induced Draft and Sturtevant Fans.

Items.	Number of Test.				
	1	2	3	4	5
Duration . . . . . . hours, minutes,	5:37	5:57	3:47	5:14	5:0
Mean boiler pressure . . . . pounds,	166	165.5	166.8	166.4	167
Mean revolutions, main engine	80.77	83.48	82.34	87.63	76.2
Mean revolutions, fan engine	265	270	267	361	248
Indicated horse-power	1,547.7	1,748.4	1,704.2	2,066.4	1,352.3
Heating surface per I. H. P. . . square feet,	4.02	3.57	3.65	3.01	4.6
I. H. P. per square foot of grate	12.9	14.57	14.2	17.22	11.27
Speed . . . . miles per hour,	14.21	13.3	13.04	14.02	12.04
Temperature of air entering furnace, degrees,	235	285	365	390	350
Temperature of air entering fan . degrees,	——	405	530	590	450
Draft at furnace . . . . inches of water,	0.21	0.23	0.5	0.25	——
Draft at fan . . . . inches of water,	——	1.55	1.65	2.5	——
Coal burned per hour . . . . pounds,	2,715	3,376	3,711	3,546	——
Combustible per hour . . . . pounds,	2,370	2,947	3,240	3,095	——
Coal per I. H. P. per hour . . . pounds,	1.75	1.92	2.17	1.71	——
Coal per square foot of grate per hour, pounds,	22.6	28.13	30.9	29.55	——
Coal per ton of cargo, per mile . pounds,	——	0.063	0.071	0.063	——
Water evaporated per hour . . . pounds,	24,976	29,782	30,129	33,990	22,682
Water per I. H. P. per hour . . . pounds,	16.13	17.03	17.69	16.45	16.69
Water per pound of coal . . . pounds,	9.2	8.82	8.18	9.52	——
Water per pound of combustible . pounds,	10.53	10.10	9.29	10.98	——
Water from and at 212° per pound of combustible, pounds,	11.45	10.91	10.12	11.97	——

"In this particular case the fans are placed, one at the back end of each boiler and against the bulkhead between the engine and the boiler rooms." [The arrangement is similar to that shown in Fig. 56, in connection with which the general features of the system are presented. The general results of the test are given in Table No. 127.]

"Test No. 1 was made on Lake Huron, going up, with vessel drawing 13 feet 8 inches aft and 6 feet forward, compartments aft full and those forward 3 feet deep. . . . The vessel loaded 132,500 bushels of wheat at Duluth, equivalent to 3,975 net tons, and draft forward and aft was 14 feet 5 inches and 14 feet 9 inches respectively. Test No. 2 was made on Lake Superior, coming down. Electric engine was running. Test No. 3 was made on Lake Huron, coming down, during daylight. Dampers in ashpit were closed. Test No. 4 was made on Lake Huron, in daylight. Dampers were open. Test No. 5 was made at night. Dampers were closed. . . . Weighed all ashes made during Test No. 2 and found them to amount to 12.7 per cent."

Gordon Hollow Blast Grate.—One of the devices for equably distributing the air and stimulating combustion in connection with mechanical draft is the hollow grate bar. The particular apparatus here described, of which a Sturtevant blower forms an inherent part, [1]"may be resolved into three principal parts,—the grate bars themselves, lying just where grate bars usually do; the main blast pipe, extending from just outside the furnace wall transversely through the furnace, at right angles to the grate bars, or the ground line, just in advance of the bridge wall; and the connecting tubes, extending vertically from the main blast pipe below to the grate bars above, and establishing connection between the two."

"The blast-grate bars are of two kinds: 1. The Combination Draft and Blast Grate bar for coal, coal refuse, coke, bagasse, tan bark, etc.; 2. The Tuyere grate bar for wood, sawdust and wood refuse generally."

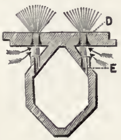

FIG. 57.

"The Combination Draft and Blast Grate bars . . . may be briefly described as consisting of a perforated top [shown in general view, Fig. 59] and a hollow lozenge-shaped body, whose laterally projecting angles are provided with series of orifices, E, below and registering with the perforations, D, in the top of the grate bar [see Fig. 57]. . . . When the blast pressure is removed, the unobstructed draft perforation, D, in the top of the grate performs its useful function, and the natural draft prevails."

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[1] The Gordon Patent Hollow Blast Grate, Catalogue, 16 pp. Gordon Hollow Blast Grate Co., Greenville, Mich.

"The Tuyere grate may be described as follows: An air chamber or duct having a transverse area 3½ x 6 inches extends through the body of the grate bar. In the heavy top of the bar four or five flaring openings, each 7 inches in diameter, are cast. These openings are afterwards bored and reamed to a uniform size for the reception of the heavy lid-shaped valve with which each is fitted. As shown in the accompanying cut [Fig. 58], these valves are each provided with a series of circumferential notches through which the air within the bar escapes to fan the furnace fire."

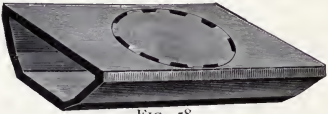

Fig. 58.

"Instead of being laid side by side in the furnace like the Combination Draft and Blast Grate bars, the Tuyere grate bars are alternated with perforated draft-grate bars of special design, each of which is likewise 8 inches wide, and of course of the same length as the Tuyere grate bars themselves. The reason for this difference in the arrangement of the Combination Draft and Blast Grate bars and the Tuyere grate bars is, that while the former bar combines in itself the functions of both draft and blast-grate bar, the latter is primarily designed as a blast-grate bar only, so that the intermediate draft-grate bar is required to supply the natural draft."

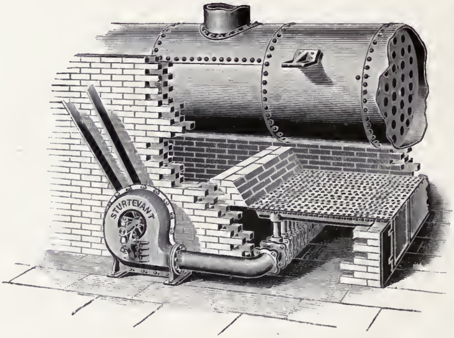

Fig. 59. Application of Sturtevant Fan to Gordon Hollow Blast Grates.

"To burn well a fire must have a sufficiency of air. Nor that alone. To be effective, the air must thoroughly intermingle with the fuel particles. It is because of this necessity that natural draft is so inadequate, and the Gordon patent hollow blast-grate system so satisfactory where sawdust, coal dust and similar fuels are to be burned. Whereas with the former the air can come into contact with the densely packed mass of sawdust or coal dust only on the surface, so the bed of fuel burns on top alone, and at the edges; with the latter, the air is, through hundreds of holes, forced with firm yet violent pressure through the mass, which it thus loosens and thoroughly permeates, so that everywhere are found dust and air in intimate contact,—a condition most favorable to perfect combustion."

A positive supply of air under considerable pressure has been found to be absolutely essential to the successful burning of bagasse. In Fig. 60 is illustrated a form of bagasse burner in which the Gordon hollow blast grates just described are employed in connection with a Sturtevant blower. The bagasse is brought to the boilers by means of a carrier, whence it is delivered to the furnaces through hoppers, as shown. The regular supply of fuel, the freedom from admission of cold air through frequently opened fire doors, the tuyere form of the grates and the constant and absolute supply of air under pressure combine to produce the most intense heat. The fire-brick arch above the grates prevents waste of heat, radiates it back upon the fire and thereby plays a most important part in maintaining a high temperature within the furnace chamber.

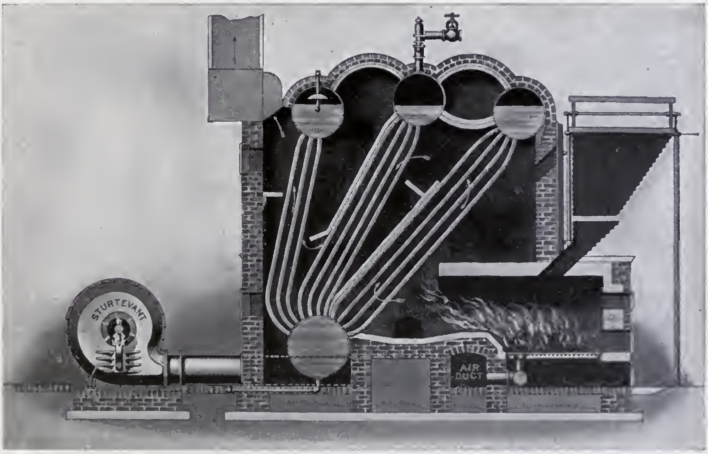

FIG. 60. BAGASSE BURNER WITH GORDON HOLLOW BLAST GRATES AND STURTEVANT BLOWER.

Gadey Air Grate. — This is another form of hollow blast grate in connection with which the Sturtevant fans are employed. A general view of a boiler equipped with these grates and a Sturtevant blower is presented in Fig. 61.

[1] "The Gadey air grate is composed of hollow cast-iron grate bars 2¾ inches wide, so constructed that when they are placed together to form a grate, a uni-

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[1] The Gadey Air Grate, Catalogue, 16 pages, 1896. Brown Brothers Manufacturing Company, Chicago, Ill.

form supply of air can be injected into them from a pressure blower and delivered from the interior of the bars through slots to the surface of the grate. This compressed air, having its outlet at the surface of the grates and siphoning the natural draft between the bars, distributes its oxygen to every part of the fuel on the grate and creates rapid and perfect combustion."

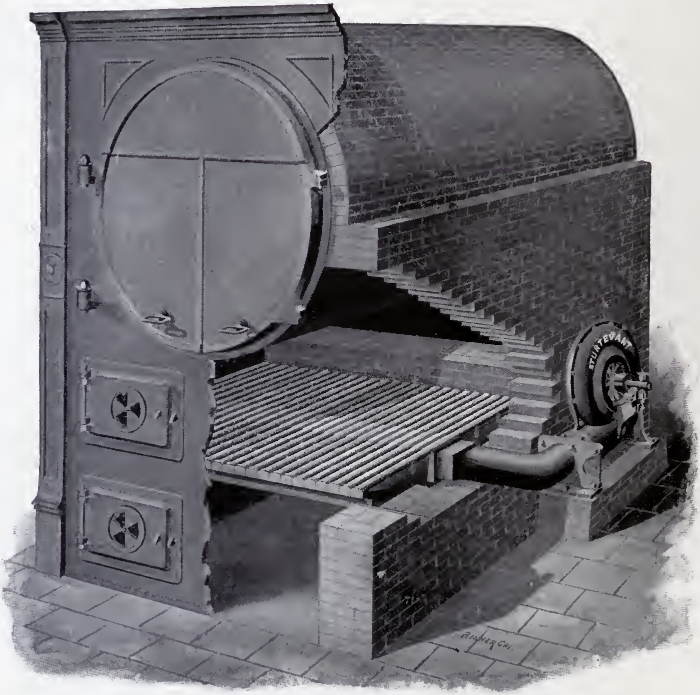

Fig. 61. Boiler Equipped with Gadey Air Grate and Sturtevant Blower.

"In no other way can complete combustion be effected than by the thorough penetration of oxygen through the mass of fuel on the grates. To effect this result with the use of air grates, the apertures or vents through which the air is delivered to the fuel should be as close together as possible. In the Gadey air grate these vents are 2½ inches apart and extend longitudinally almost the entire length of the bar.

The cross-sectional view, Fig. 62, "shows the position of the slots or air vents in the side of the bars near the surface. It will be observed that the air is forced through the slot in an oblique direction, or at an angle of 45 degrees. When the two currents of air from adjoining bars meet they create a suction or siphonage through the natural draft openings between the bars, thus giving an increased volume of air to the fuel. The outer edges of the bars overhanging the slot prevent ashes or clinkers from falling into the interior of the bars and form a drip for slag and ashes to drop into the ashpit.

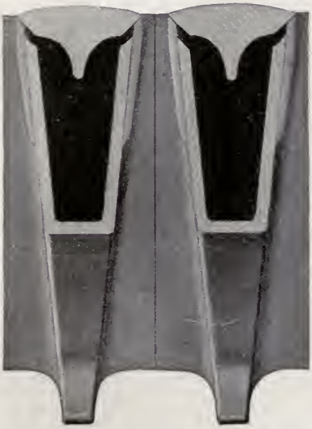

Fig. 62. Section of Gadey Air-Grate Bars.

"The Gadey air grate is especially adapted to the burning of screenings, sawdust, bagasse or any form of refuse which, from its fineness or tendency to pack closely on the grates, is difficult, and in fact impossible, to burn successfully in any other kind of furnace."

Cheney Brothers, South Manchester, Conn.[1] — "The silk mills of Messrs. Cheney Brothers at South Manchester, Conn., have been in active operation for the past fifty years, and at the present time offer employment to about 2,000 persons engaged in the spinning, weaving, dyeing and printing of silk and velvet goods.

"The steam plant as a whole is not a new one, but one which has been extended and improved from time to time to meet the growth of the business. It is a plant, however, in which the owners fully recognized the advantages of modern labor-saving machinery and the devices that tend to increase the economical generation of power About two years ago the old boiler house at the lower mills was rebuilt, and new boilers, coal-handling machinery, a flue heater or economizer, and exhaust fan for handling the products of combustion from the boilers, were installed."

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[1] Boiler Plant of Cheney Bros.' Silk Mills. The Engineering Record, New York, Jan. 6, 1894.

The general arrangement of the induced draft, as designed, is presented in Fig. 63. The fan is a special 7 x 10½ Sturtevant exhauster, so arranged in connection with flue dampers that the gases may be passed through the economizer and the fan, as is usually the case in order to create sufficient draft, or direct to the chimney. The fan was designed with a special three-pulley arrangement, so that it could be driven either by the independent Sturtevant engine or by belt from the line shaft.

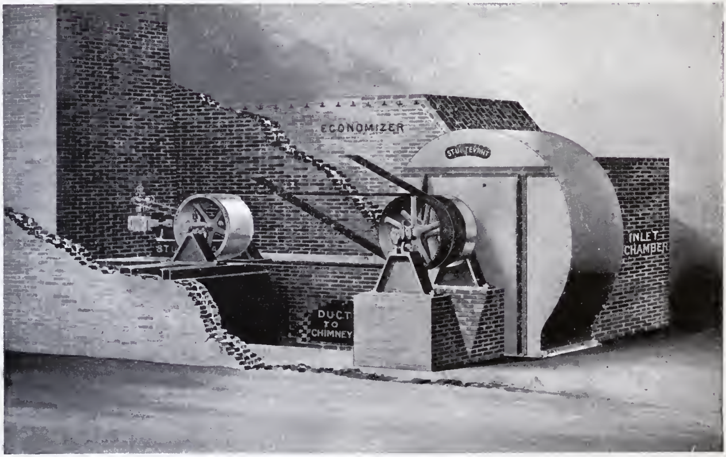

FIG. 63. INDUCED-DRAFT PLANT AS DESIGNED FOR CHENEY BROTHERS' SILK MILLS, SOUTH MANCHESTER, CONN.

"The lower mills comprise buildings of a combined area of 110,000 square feet, whose purposes require a large amount of steam for lighting, power, heating and dyeing. For all these purposes steam at 55 pounds pressure is generated in four Babcock & Wilcox boilers of 1,000 horse-power. Live steam is used in ceiling coils in cold weather to heat the buildings, while reducing valves are used to throttle the steam employed in the dyeing cylinders."

"The economizer contains 480 pipes 5 inches in diameter and 10 feet long, presenting about 6,000 square feet of heating surface against 12,000 square feet in the boilers. As usually run, the economizer not only heats the water to about 212° Fahr., but supplies about 50 gallons per minute at the same temperature to the dyehouse. An average of hourly readings during one day, of the temperatures of water and flue gases when under the above conditions, gave the following results, 750 horse-power being in use" : —

Temperature water entering Berryman heater . . .	45 degrees.
Quantity water entering Berryman heater per minute .	95 gallons.
Temperature water entering economizer	112 degrees.
Temperature water leaving economizer	211 degrees.
Temperature flue gases leaving economizer . . .	275 degrees.

Before the economizer was installed the flue gases entered the chimney at a temperature of about 475 degrees.

"The mechanical draft plant was introduced in this plant to save the construction of an expensive stack, the old stack being in good order and amply large to discharge the gases when aided by the fan."

This points clearly to the necessity of increased draft in connection with an economizer. That a fan is more desirable than a chimney under these circumstances is evidenced by the following statement:[1] "The gases are discharged through a chimney 90 feet high. This chimney we used previous to remodelling our plant, and while it is higher than we actually need to discharge the gases from the fan, we find it a satisfactory arrangement, as it will give sufficient draft to start the fires and do our night work without using the fan. As our work is very variable, we have never made any accurate tests, but feel satisfied that the fan has saved us the cost of building a new chimney at least 175 feet high, and that it is giving us as satisfactory results as we could have obtained by a chimney."

The Deringer Colliery of the Cross Creek Coal Company, Deringer, Pa. — This plant, which is illustrated in Fig. 64, consists of two Babcock & Wilcox boilers having an aggregate builder's rating of 500 horse-power, equipped with a Coxe mechanical stoker and a No. 8 Sturtevant "Monogram" blower for producing the requisite draft. Each boiler contains 4,225 square feet of water-heating surface, the aggregate grate area is 100 square feet, the free air space for passage of air amounting to 20 per cent of the full area.

"The furnace," as described in detail by Mr. Coxe,[2] and as herewith illustrated by the accompanying cut, Fig. 65, "consists essentially of a travelling grate moving from the right toward the left. The coal, which is brought to the hopper 20 by a drag, spout, or any other convenient method, feeds down by

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[1] Cheney Brothers, South Manchester, Conn. Letter of Feb. 25, 1896, to B. F. Sturtevant Co.

[2] Some Thoughts on the Economical Production of Steam, with Special Reference to the Use of Cheap Fuel, by a Miner of Coal. Eckley B. Coxe. Transactions New England Cotton Manufacturers' Association, April, 1895.

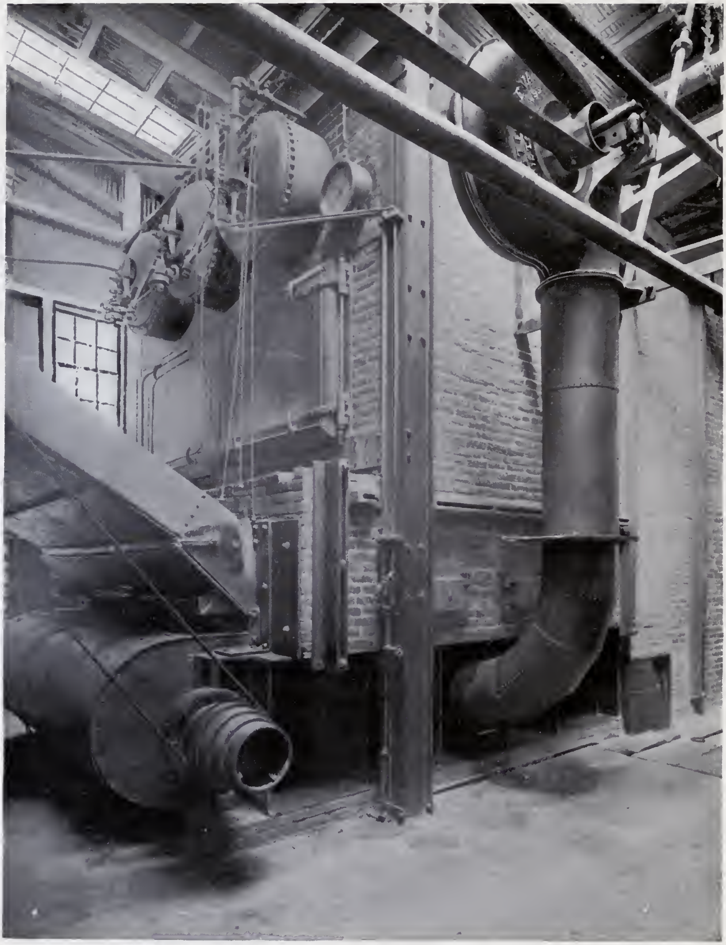

FIG. 64. ARRANGEMENT OF COXE MECHANICAL STOKER AND STURTEVANT FAN AT THE DERINGER COLLIERY OF THE CROSS CREEK COAL COMPANY, DERINGER, PA.

gravity over the fire brick 14 on to the travelling grate. The coal is carried slowly at the rate of from 3½ to 5 feet per hour toward the other end. In the beginning of the operation the coal on the right-hand side of the furnace is ignited, the other part being covered with ashes or partially consumed coal. After the furnace is heated, the fire brick 14, which we call the 'ignition brick,' becomes hot, and the coal, passing down under regulating gate 21, becomes gradually heated, and by the time it reaches the foot of the ignition brick is incandescent. In some cases the coal becomes hot enough to ignite soon after it passes the regulating gate 21. Under the grate there are a number of chambers made of sheet iron which are closed on all sides except on top. The blast from the fan which is used to furnish the air is blown into the large air chamber, which is the second one from the right. These air chambers are open on top, but the partitions are covered by plates 27, 28, 29 and 30. These plates are of such width that no matter what may be the position of the grate bars 18, there is always one resting upon this plate, so that the air cannot pass from one chamber to another except by leakage along the bar. The result of this arrangement is that if we are blowing into the large air chamber with a pressure, say, of 1-inch water gauge, the pressure in the next air chamber to the left would be about ¾ inch, the next to that ½ inch, and the next to that ¼ inch. Of course these figures are not strictly correct, and are used merely for the purpose of illustrating, as I am now describing only the general principle of the apparatus. The pressure in the air chamber to the right would be, say, ¾ inch. The result of this state of affairs is, that the coal, when it arrives on the grate, is

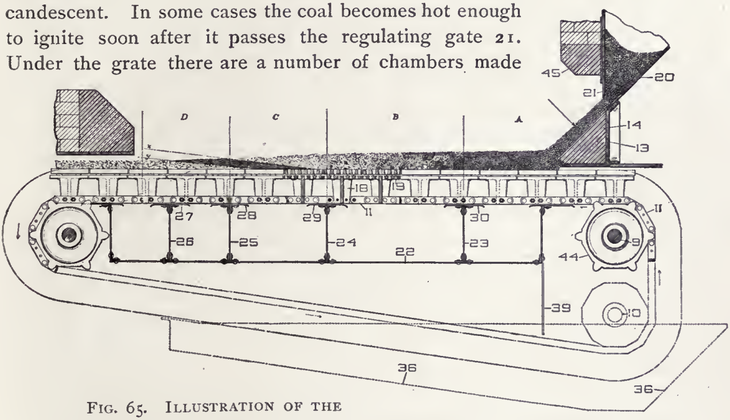

Fig. 65. Illustration of the Coxe Process and Furnace for Burning Small-Size Anthracite Coal.

subjected to a pressure of blast sufficient to ignite it, but not too strong to impede ignition. In order to regulate exactly the pressure of the air in each of the compartments, the partitions are provided with registers, by the simple opening and closing of which the pressure in the air chambers can be varied to suit the conditions.

"As the thoroughly ignited coal passes slowly over the second compartment (where the air pressure is a maximum) it burns briskly, and then slowly passes over the third compartment, where the air pressure is less and better suited to the combustion of the thinner layer of partly consumed coal. The bed continues to diminish in carbon and to be subjected to less blast, until finally the hot ashes are cooled off (before being dumped) by a very gentle current of air, which is heated and mingles with the carbonic oxide produced in the zone of intense combustion B, and converts it into carbonic acid; the object being to subject the coal, as soon as it arrives on the grate, to a pressure of blast which is the proper one to ignite it, then to burn it with a blast as strong as will produce good combustion, and, as the carbon is eliminated and the thickness of the bed becomes smaller, to diminish the blast to correspond to these conditions. The mass of coal remains all the time in practically the same position and condition in which it was placed on the grate, except so far as altered by the combustion."

"From this brief description the continuous action of the furnace can be easily understood. The coal, passing continuously down from the ignition brick, is ignited gradually, burned out, and the ashes are carried off or dumped by the grate bars as they descend and become vertical."

The general results of two careful tests of this plant are given in Table No. 128. The coal used in each case was (anthracite) rice. The results of other tests of other small anthracite coals in connection with a Coxe stoker have already been given in Table No. 46. It was for the especial purpose of successfully burning such coal that this type of furnace was designed and mechanical draft adopted. The general features of the system are thus stated by Mr. Coxe in the paper from which quotation has just been made:—

"We can blow with dry air. It is not necessary to use steam to partially prevent the formation of clinker, and consequently we avoid the loss due to heating the steam in the fire, the waste of it in producing the blast, and the effects of its decomposition. While the furnace is running regularly it produces at all times the same results; that is to say, you are always evaporating exactly the same number of pounds of water per minute, and can therefore furnish a steady supply of steam. No time is lost in cleaning fires, nor is cold air introduced when shovelling in coal, etc. If it is desired to reduce the production of steam, it is only necessary to slow down the engine so as to reduce the blast, the speed

Table No. 128.— Results of Tests of Babcock & Wilcox Boilers with Coxe Mechanical Stoker and Sturtevant Fan at The Deringer Colliery of The Cross Creek Coal Company, Deringer, Pa.

Date of test		Dec. 9, '96.	Mar. 16, '96.
Number of boilers		Two.	One.
Duration of test	hours,	10	10
Steam-gauge pressure	pounds,	110	103.8
Air pressure in ashpit	inches of water,	1.00	0.60
Temperature of feed-water	degrees Fahr.,	45	42
Temperature of escaping gases	degrees Fahr.,	900	499.5
Total dry coal consumed	pounds,	40,692.4	9,800
Total combustion	pounds,	33,290.4	8,200
Total weight of water apparently evaporated	pounds,	277,863	75,956
Equivalent water evaporated into dry steam from and at 212°,	pounds,	276,853	92,514
Equivalent water evaporated per pound of dry coal from and at 212°,	pounds,	6.83	9.44
Equivalent water evaporated per pound of combustible from and at 212°,	pounds,	8.51	11.28
Water evaporated per hour per square foot of grate surface,	pounds,	276.8	92.5
Dry coal burned per square foot of grate per hour	pounds,	20.34	9.8
Horse-power on basis of 30 pounds of water evaporated per hour from temperature of 100° into dry steam of 70 pounds gauge pressure,	horse-power,	802.5	268.1
Per cent of horse-power developed above rating		60.5	7.25

of the grate, and, if it is desired, the thickness of the bed of fuel can be changed. By means of the registers between the different air compartments we are able to regulate the supply of air in all parts of the bed of fuel so as to get the best results, and to control to a large extent the composition of our stack gases. Without difficulty we can obtain over 16 per cent of carbonic acid and less than 3 per cent of free oxygen without carbonic oxide, a result hardly possible with hand firing; at least we have not been able to find such percentages in any of the published reports of tests. The same thing applies to the ash. We can, by regulating the thickness of the bed, the speed of the grate, and the distribution of the air, reduce the ash as low as is probably economical. We have obtained ash very low in carbon, but we think it was at the expense of capacity, and probably also of heat in the stack gases — too much air being introduced. We can certainly burn the finer coals with a less excess of air than we hoped for or than is generally done. We have shown that the amount of water evaporated per pound of coal does not depend practically upon the size, but only upon the amount of heat units or combustible in the coal. We have found that we could use coal very high in ash without diminishing very materially the number of pounds of water evaporated per pound of combustible; but we have also found that the purer the coal and the larger the size, the greater is the capacity of the boiler. That is to say, that we can evaporate more water with a given boiler, the purer the coal and the larger the size; although the quantity of water evaporated per pound of combustible is practically the same in all cases, so long as there is no great excess of dust in the fuel, which in that case stops up the air passages between the pieces of coal and thereby prevents a regular blast and even burning of the fuel. We have also found that we could burn bituminous coal without much difficulty, and by properly regulating the air avoid absolutely the production of smoke. We think that by providing sufficient grate surface we can evaporate as much water, or nearly as much, with a ton of No. 3 buckwheat or rice coal as we can with pea coal, provided they are equally free from impurities; the only additional expense in the case of the small coal being the interest and depreciation on the additional plant necessary to produce a given amount of steam."

Central Unidad, Cuba.— This is a bagasse-burner plant consisting of two batteries of two Babcock & Wilcox boilers, each battery of an aggregate of 856 rated horse-power, being equipped with a Cook hot-blast green-bagasse burner and a No. 10 Sturtevant "Monogram" fan. A view of one of the batteries is presented in Fig. 66, while the results of a careful test are given in Table No. 129.

Bagasse usually contains enough sugar, if used as fuel, to evaporate the contained water. If, therefore, it can be burned direct from the mill without the loss of the sugar due to sun drying, it should give as good results as when dried.

[1] "Cook's automatic apparatus accomplishes this result, burning the bagasse automatically direct from the sugar mill, with a saving of the large number of men, carts and oxen required for spreading, drying, gathering and firing it in a dry state. It also secures far better combustion than can be had with the best hand firing, with no smoke, little refuse and a greatly increased evaporative capacity. An element of additional economy consists in utilizing the waste heat escaping to the chimney for heating the blast. This hot blast is peculiarly efficient in burning wet fuel, because of the greatly increased capacity of the hot air for absorbing moisture, and thus partially drying the bagasse before burning. . . . These considerations explain the fact that where these burners have been erected they have always brought about a large reduction in the supplementary fuel required with dry bagasse, besides giving more and steadier steam pressure. In a well-arranged plantation the bagasse is sufficient without other fuel."

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[1] Steam. Catalogue, 1897. Babcock & Wilcox Company, New York.

"The furnace of Cook's apparatus consists in an oven of brick, having a smaller chamber beneath, into which the blast previously heated is introduced through numerous perforations in the walls. Openings in the walls of the oven permit the escape of the gases of combustion to the boilers. On their way to the chimney these gases pass tubular heaters, through which a fan forces the blast *en route* to the burner, thus retaining a large part of the waste heat to the furnace and securing an exceedingly high temperature therein. The furnaces require to be cleaned once in 24 hours, when the refuse from 250 tons of bagasse makes about four wheelbarrow loads, in the form of a vitreous mass, evidencing the intense heat attained.

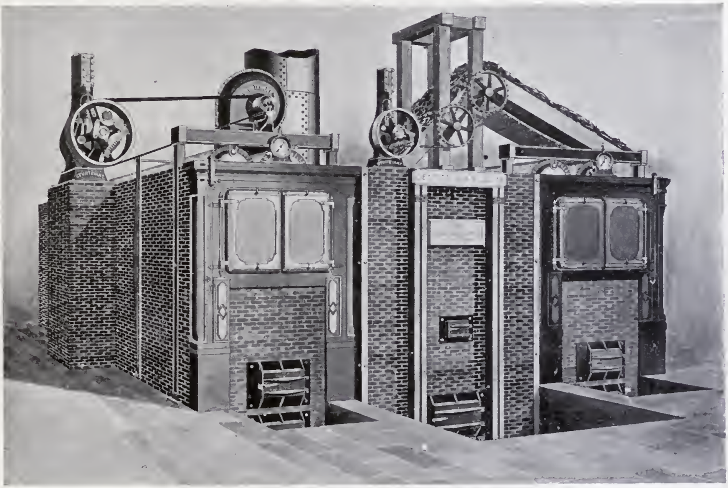

FIG. 66. COOK'S BAGASSE BURNER WITH STURTEVANT FAN, AT CENTRAL UNIDAD, CUBA.

"The bagasse is fed to the furnaces automatically by an arrangement of carriers which receive it from the rolls and distribute it equably to the different furnaces, where more than one is required, dumping any surplus upon cars, where it is stored for use when the mill is not grinding."

The fan, as shown in the illustration, is driven by a Sturtevant upright engine, which thus renders the draft entirely independent of climatic conditions and of any other means of production.

Table No. 129. — Results of Test of Cook's Patent Hot-Blast Green-Bagasse Burner with Sturtevant Fan, at Central Unidad, Cuba.

Duration of test	hours,	9.75
Heating surface in boilers	square feet,	9,828
Average temperature of feed-water	degrees Fahr.,	169.35
Average temperature of air in heater chambers	degrees Fahr.,	505
Average temperature of air in tuyere chambers	degrees Fahr.,	233
Average temperature of air in blast boxes	degrees Fahr.,	235
Average temperature of atmosphere	degrees Fahr.,	82
Average temperature of atmosphere over heaters	degrees Fahr.,	92
Average temperature of gases at base of stack	degrees Fahr.,	468
Total pounds of cane ground		745,290
Pounds of cane ground per hour		76,440
Pounds of cane ground to produce bagasse burned per hour		65,532
Pounds of cane ground to produce bagasse for 1 horse-power per hour		44.82
Total pounds of bagasse made		221,133
Pounds of bagasse made per hour		22,680
Total pounds of bagasse burned during test		189,633
Pounds of bagasse burned per hour		19,449
Total pounds of bagasse not burned, or spare		31,500
Pounds of bagasse not burned, or spare, per hour		3,230
Pounds of bagasse burned to develop 1 horse-power, per hour		13.3
Pounds of juice extracted		524,157
Average density of juice	degrees Baumè,	10.6
Per cent extraction		70.32
Per cent bagasse		29.68
Average steam pressure	pounds,	87
Average air pressure under blowers	inches water,	2.15
Average air pressure in tuyere chambers	inches water,	1.55
Average air pressure in blast boxes	inches water,	1.6
Average draft current in stack base	inches water,	.35
Per cent dry solids in 100 pounds bagasse		53.5
Per cent moisture in 100 pounds bagasse		46.5
Per cent dry solids in 100 pounds cane		15.87
Total pounds of water evaporated (tank measurement)		408,540
Pounds of water evaporated per hour		41,901
Pounds of water evaporated per square foot of heating surface per hour		4.26
Pounds of water evaporated per one pound of bagasse burned		2.154
Pounds of water evaporated from and at 212° per one pound of bagasse burned,		2.328
Pounds of water evaporated from 212° into steam of 70 pounds pressure per one pound of bagasse burned,		2.245
Rated horse-power of boilers		856
Horse-power of boilers developed (30 pounds of water evaporated from 212° to steam of 70 pounds pressure),		1,462
Per cent above rated capacity		70

Table 129. — Concluded.

Horse-power per hour not used in the spare bagasse	242
Square feet of heating surface which developed 1 horse-power per hour	6.72
Quality of steam	Dry.
Per cent of ash on per cent of bagasse	0.78
Average revolutions per minute of blower engine	152
Average revolutions per minute of blowers	968
Temperature of bagasse burner at 7 feet from hearth, indicated by melting of copper rod 3/8 inch diameter, degrees Fahr.,	2,548

L. B. Darling Fertilizer Company, Pawtucket, R. I. — A good illustration of the use of under-grate forced draft in the combustion of small anthracite coal without special appliances other than the introduction of a fan for producing the requisite draft is shown by the experience of this concern. The boilers, which are of the horizontal return tubular type, set in a single battery, all supplied by the same fan, have an aggregate nominal rating of 450 horse-power and a combined grate area of 121 square feet.

As stated in a recent communication, [1] "the coal used is buckwheat, clear, and costs $2.87 per ton of 2,000 pounds. The application of the forced draft is through iron ducts over the top of the boilers, with iron ducts dropping to each boiler at the back end, and running through the combustion chamber, to and through the bridge wall to the ashpit. In the bridge wall is set a blast box 54 inches by 4 inches; this box is made with a door, hinged at the top to swing to regulate the flow of air to each boiler. Under ordinary conditions the rate of combustion is about 10 pounds per square foot of grate per hour. We find the temperature of the flue gases to be 150° Fahr. lower when the blower is in operation than when it is not in operation or with natural draft."

This plant is equipped with a special Sturtevant steel-plate steam fan with direct-connected double upright engine, whose speed is automatically controlled by a Burke regulating device, by means of which the speed is increased as the steam pressure slightly decreases, and *vice versa*, so that a practically constant steam pressure is maintained.

The fan is located between one end of the battery of boilers and the wall of the building, occupies no valuable floor area and discharges the air directly upward into the horizontal iron duct above the boilers. This arrangement illustrates the feasibility of introducing such a system of draft production in an old boiler plant.

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[1] L. B. Darling Fertilizer Company, Pawtucket, R. I. Letter of Oct. 13, 1897, to B. F. Sturtevant Co.

Cleveland Iron Mining Company, Ishpeming, Mich. — This plant serves as an excellent illustration of the application of a Sturtevant blower in connection with an under-feed mechanical stoker, a type of stoker in the operation of which considerable draft pressure, such as can best be produced by a blower, is an absolute necessity. This plant, shown in Fig. 67, consists of a battery of 72-inch return tubular boilers, equipped with Jones under-feed mechanical stokers, No. 7 Sturtevant "Monogram" blower and a Sturtevant upright engine.

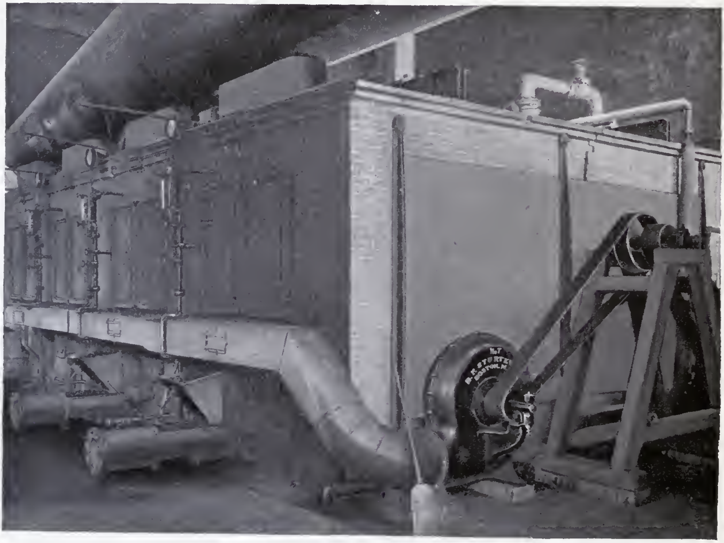

FIG. 67. ARRANGEMENT OF JONES UNDER-FEED MECHANICAL STOKERS AND STURTEVANT FAN AT CLEVELAND IRON MINING COMPANY, ISHPEMING, MICH.

[1] "The stoker consists of a steam cylinder or ram, with hopper for holding the coal, outside the furnace proper, and a retort or fuel magazine inside the furnace, into which the green fuel is forced by means of the ram; tuyere blocks, for the admission of air, being placed on either side thereof; the retort containing at its lowest point, and at a point where the fire never reaches, an auxiliary

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[1] The Improved Jones Under-Feed Mechanical Stoker. Catalogue, 32 pp., 1896. The Weeks-Eldred Company, Toronto, Ont.

ram or 'pusher,' by means of which an even distribution of the coal is obtained.

"By means of the rams, coal is forced underneath the fire, each charge of fuel raising the preceding charge upward, until it reaches the fire; which point it does not reach until it has been thoroughly coked. When in its coked state it is forced upward into the fire. The gases being liberated under the fire, and at that point mixed with air, must necessarily pass through the fire and be consumed; thus giving the benefit of all combustible matter in the fuel. Air is forced, at a low pressure, through the tuyere blocks, under the burning fuel, by means of a blower, operated by an independent engine, or from a line shaft, if such arrangement can be made."

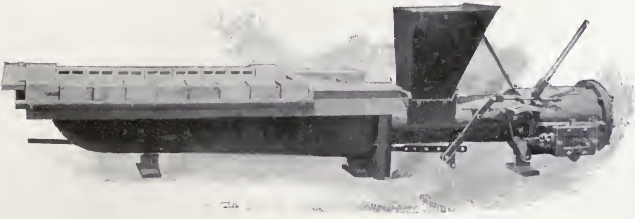

Fig. 68. Jones Under-Feed Stoker.

The stoker is shown in Fig. 68, while Fig. 69 indicates the manner of forcing upward and distributing the fuel, and the method of introducing air, which in this type of stoker is supplied by a Sturtevant "Monogram" blower.

"Coal being in the hopper, and the ram plunger at its forward stroke, when more coal is needed the ram plunger is shifted by moving the lever; coal then falls in front of the plunger and upon return movement is forced into the retort; this movement being repeated until sufficient fuel is in the retort. . . . Air at low pressure being admitted into the air chamber and through the tuyere blocks, over the top of the green fuel in the retort, but *under and through the burning fuel*, the result is that the heat from the burning fuel

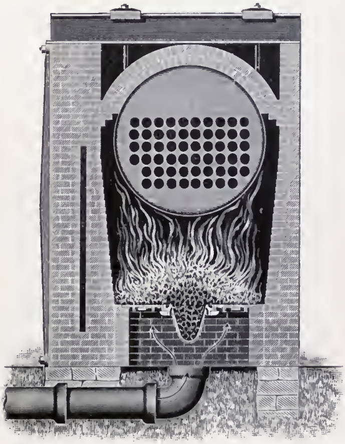

Fig. 69. Cross-Section Showing Furnace in Operation.

over the retort slowly liberates the gas from the green fuel in the retort. This gas, being thoroughly mixed with the incoming air before it passes through the burning fuel above, results in a bright, clear fire, *free from smoke*, and the complete consumption of all the heat-producing elements in the fuel. The retort being air-tight from below, and the fuel being in a compact mass in the retort, the air will find its way in the direction of the least resistance, which is upward, consequently combustion takes place only *above* the air slots; hence the castings of the retort are always cool and not subject to the action of the fire.

Table No. 130.—Results of Comparative Tests of Jones Under-Feed Mechanical Stoker Equipped with Sturtevant Blower, and Hand-Firing, at Cleveland Iron Mining Company, Ishpeming, Mich.

ITEMS.		Hand-Fired.	Stoker-Fired.
Duration of test	continuous hours,	72	72
Average steam pressure	pounds,	118.95	118.95
Average temperature of feed-water	degrees Fahr.,	210	41.1
Total coal consumed	pounds,	50,600	54,864
Total ash and clinker	pounds,	8,245	8,202.5
Total combustible	pounds,	42,354	46,661.5
Per cent of ash and clinker		16.29	14.9
Total water evaporated	pounds,	312,323	341,483
Total water evaporated per pound of coal at observed temperature,	pounds,	6,176.17	6,226.22
Water evaporated per pound of coal from and at 212°,	pounds,	6.45	7.57
Water evaporated per pound of combustible from and at 212°,	pounds,	7.71	8.91
Gain in evaporation	per cent,		17.37

The incoming fresh fuel from the retort forces the resulting ash and clinker over the top of the tuyere blocks on to the side plates, from whence they may be removed at any time without in the least interfering with the fire in the centre of the furnace, resulting in a high, even temperature at all times."

In Table No. 130 are given the results of an extended comparative test, made Oct. 30, 31, Nov. 1 and 2, 1895, between this stoker and hand-firing at the Cleveland Iron Mining Company, which serve to show its relative economy. In each case the test was made upon two boilers, and the same kind of coal—bituminous slack—was used. The coal consumed in the stoker-fired boilers includes the amount used in a fifth boiler to make steam to run the blower engine. The water evaporated in the fifth boiler is not included.

John Brown & Co., Ltd., Sheffield, Eng. — The boiler to which the following description relates is one of several to which Sturtevant fans are applied at the works of the above-named firm. It is of novel construction, designed for the purpose of experimentally demonstrating the merits of the Eaves helical induced-draft system, which has been subjected to long-continued and very careful investigation by the above-named firm.

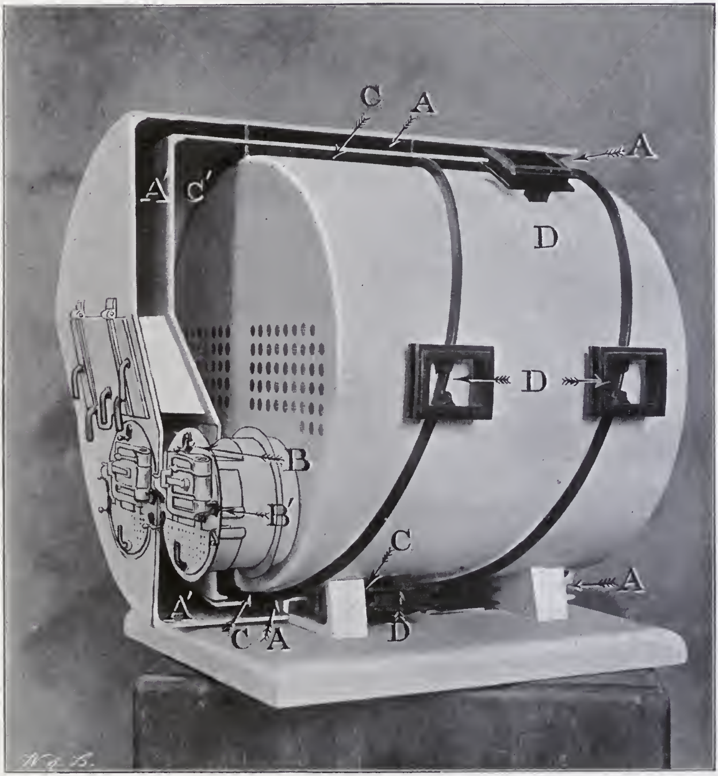

Fig. 70. Eaves Helical Induced-Draft Boiler.

This system is a combination of mechanical draft, Serve tubes and retarders and a means of abstracting a portion of the heat from the gases and utilizing it in raising the temperature of the air supplied to the furnaces. In connection with Fig. 70 the following description[1] will serve to make the system clear: —

"The cold air for the combustion of the fuel enters from the back end of the boiler, passing along the outer space A and A^1 to the valves B and B^1 in furnace fronts; on its way this cold air is guided round the outside of the 'inner' space C in a 'helical' direction by partitions set up as shown. After combustion, the waste hot gases, leaving the boiler, pass through the smoke-box into inner space C and are made by similar partitions to pass round and in close contact with boiler in a 'helical' direction on their way to the suction fan.

"The boiler by these means is thoroughly enveloped in the escaping heat, effectually preventing either radiation, condensation or straining of the boiler under any forced conditions, such as rapid generation of steam from cold water or sudden and greatly increased evaporation. The cold air on its way to the valves also absorbs a large amount of heat from the escaping gases and so enters the furnaces at a greatly increased temperature, with resultant economy.

"No blocking up of the bottom boiler tubes through any deposit in the smoke-box can take place, as such deposit, if any, drops to the bottom of inner casing C, from whence it is easily removed by doors at front. The doors D are placed so as to allow of a brush being passed through, to sweep away any sooty deposit from the inner boiler shell, should any such deposit take place.

"Referring to the annexed trials [see Table No. 131. This boiler was equipped with a Sturtevant fan with special cooling device to permit of the handling of the high-temperature gases], we find that the boiler efficiency in one case was 82 per cent and in the other 78 per cent of the actual calorific value of the coal used. If we take the mean of these figures,— namely, 80 per cent,— and work out the evaporation on the basis of best Welsh coal, we obtain the following remarkable results: —

Heat units from combustion of one pound of best Welsh coal	15,629
Latent heat of evaporation from and at 212° . .	966
Calorific value of coal in pounds of water evaporated per pound of coal from and at 212°, $\frac{15,629}{966}$ =	16.18
80 per cent of above calorific value, $\frac{16.18 \times 80}{100}$ =	12.95 pounds.

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[1] Eaves Helical Induced Draft. Catalogue, October, 1896. John Brown & Co., Ltd., Sheffield, Eng.

"Or practically the evaporation of 13 pounds of water per pound of coal from and at 212° Fahr., with a rate of combustion of over 30 pounds of coal per square foot of grate and a ratio of heating surface to grate of only 28 to 1.

Table No. 131. — Results of Test of Eaves' Helical Draft Boiler with Sturtevant Fan.

ITEMS.	First Trial.	Second Trial.
Duration . . . hours,	7	7
Total coal burned . . . pounds,	7,581	8,022
Total water evaporated . . . pounds,	71,000	71,500
Temperature of feed-water . . . degrees Fahr.,	54	50
Steam pressure . . . pounds per square inch,	43.4	45
Revolutions of fan engine per minute . . .	508	520
Temperature of air at side valve . . . degrees Fahr.,	234	259.5
Temperature of gases at inlet of fan . . . degrees Fahr.,	309	353.8
Temperature of gases in smoke box . . .	Melted bismuth, not lead.	Melted bismuth, not lead.
Vacuum under grate bars . . . inches of water,	0.75	0.64
Vacuum over fires . . . inches of water,	0.82	0.81
Vacuum at base of chimney . . . inches of water,	4.59	4.59
Vacuum above fan outlet . . . inches of water,	0.38	0.38
Velocity of air per minute under grate bars . . . feet,	1,476	1,362
Velocity of air per minute through outer casing . . . feet,	257	227
Temperature of the air entering outer casing, degrees Fahr.,	78	67
Coal burned per square foot of grate per hour, pounds,	33.84	35.81
Water evaporated per pound of coal . . . pounds,	9.36	8.91
Water evaporated per pound of coal from and at 212°, pounds,	11.12	10.63
Calorific value of coal used, expressed in pounds water evaporated from and at 212°,	13.6	13.6
Efficiency of boiler . . . per cent,	82	78.3

"To obtain an equally high evaporative efficiency with a Lancashire boiler, it would require to be fitted with an economizer having a combined total ratio of heating surface to grate surface of 75 to 1, with a rate of combustion of only 12 pounds of coal per square foot of grate. Therefore, to produce the same amount of steam per hour, — namely, 10,200 pounds, — two Lancashire boilers would be required, 8 feet in diameter by 30 feet long, with their attendant economizer, the floor space occupied being four times that required for the helical draft boiler.

"An ordinary marine boiler of the same dimensions as the one used in these trials — namely, 10 feet 6 inches diameter, by 10 feet 6 inches long — will with good natural draft evaporate about 5,000 pounds of water per hour; the efficiency being about 65 per cent, or equal to 10.5 pounds of water evaporated per pound of best Welsh coal from and at 212°, instead of 13 pounds. From this it follows that two boilers with natural draft would be required to do the work of one of same size fitted with the helical draft and 'Serve' tubes, the relative efficiency being as 65 to 80.

"In addition to the saving in space effected by this system of draft when applied to marine boilers, the saving in weight is also very great, when we consider that by the addition of about 6 tons to the weight of a boiler installation of the size under consideration — to cover the weight of the fan and engine, helical casing, and greater weight of Serve tubes — the power of the boiler can be doubled, at the same time effecting a saving in coal consumption compared with two ordinary boilers of about 2½ tons per 24 hours."

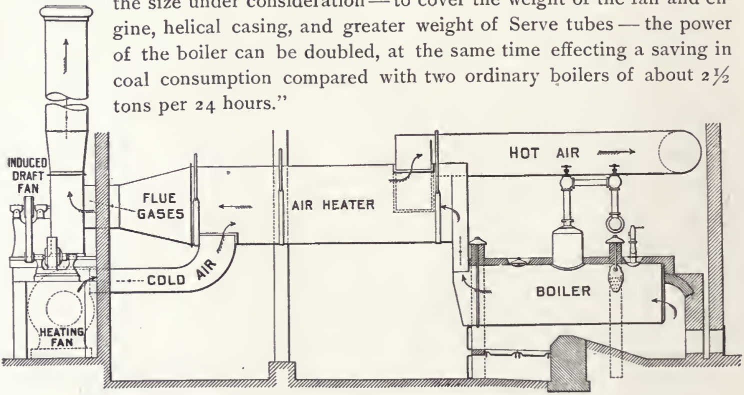

Fig. 71. Arrangement of Sturtevant Fans for Mechanical Draft and Air Heating at Union Traction Company, Philadelphia, Pa.

Union Traction Company, Philadelphia, Pa. — The complete boiler plant consists of three horizontal return tubular boilers 60 inches in diameter by 17 feet 8 inches long over all, and is employed for the generation of steam at about 15 pounds pressure for heating the car sheds, shops and offices covering an entire square. For utilizing the waste heat in the flue gases, an air heater is interposed between the boiler uptake and the stack. Through the longitudinal tubes of this heater the hot gases pass; while air admitted to the bottom of the cylindrical shell circulates around the tubes and is discharged at the top. To secure the necessary draft to overcome the resistance of the air heater, and at the same time render the draft positive and independent of climatic conditions, a 100-inch Sturtevant fan is provided, through which the gases are drawn and discharged into the vertical stack extending from the outlet of the fan. This fan is driven by belt from an independent electric motor.

To secure a positive and sufficient movement of air through the heater, another Sturtevant fan — a 70-inch steel-plate exhauster — is employed. This is driven by a direct-connected Sturtevant electric motor. Both fans are clearly shown in the view presented in Fig. 72, while the general arrangement of the entire plant is indicated in Fig. 71. Economy of space and avoidance of in-

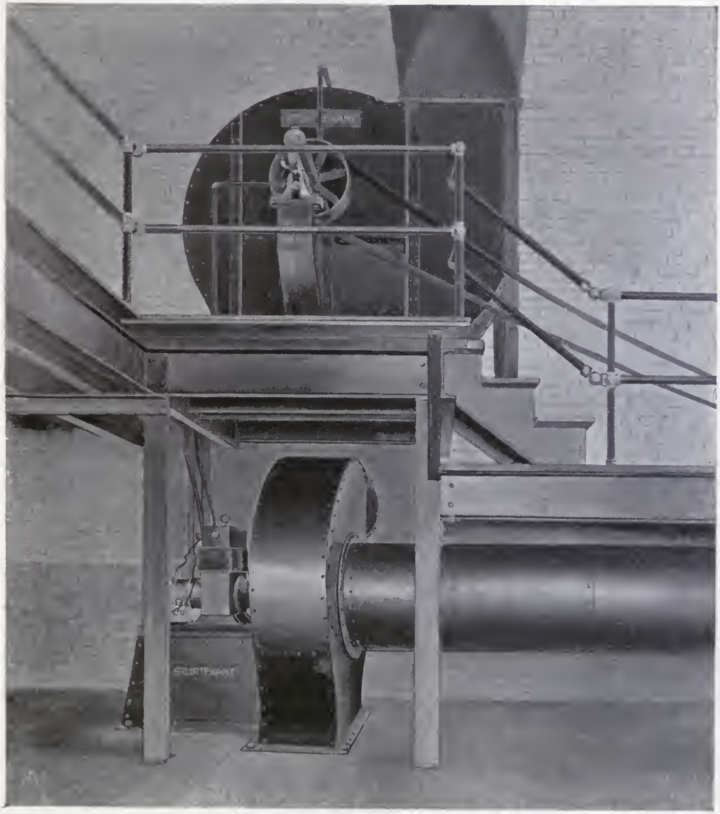

FIG. 72. STURTEVANT FANS FOR MECHANICAL DRAFT AND AIR HEATING AT UNION TRACTION COMPANY, PHILADELPHIA, PA.

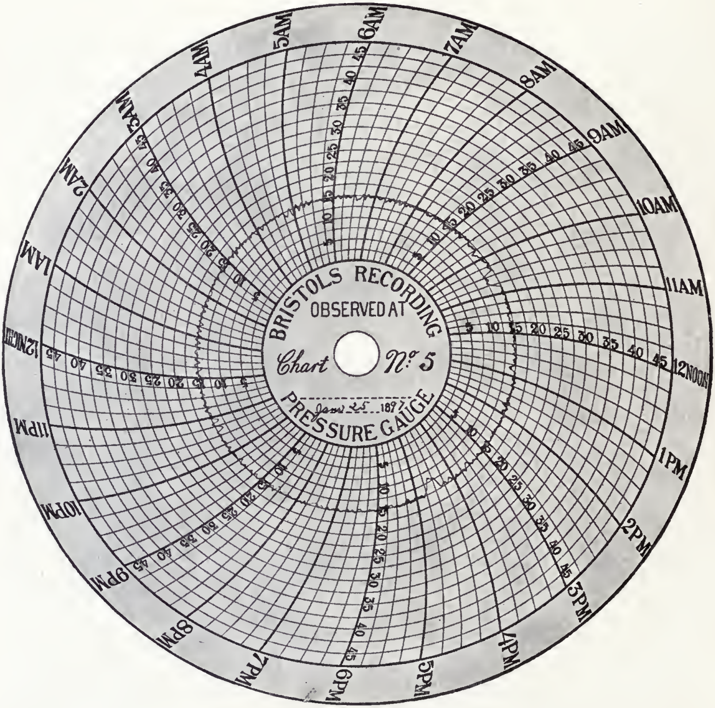

Fig. 73. Steam-Pressure Chart from Induced-Draft Plant at Union Traction Company, Philadelphia, Pa.

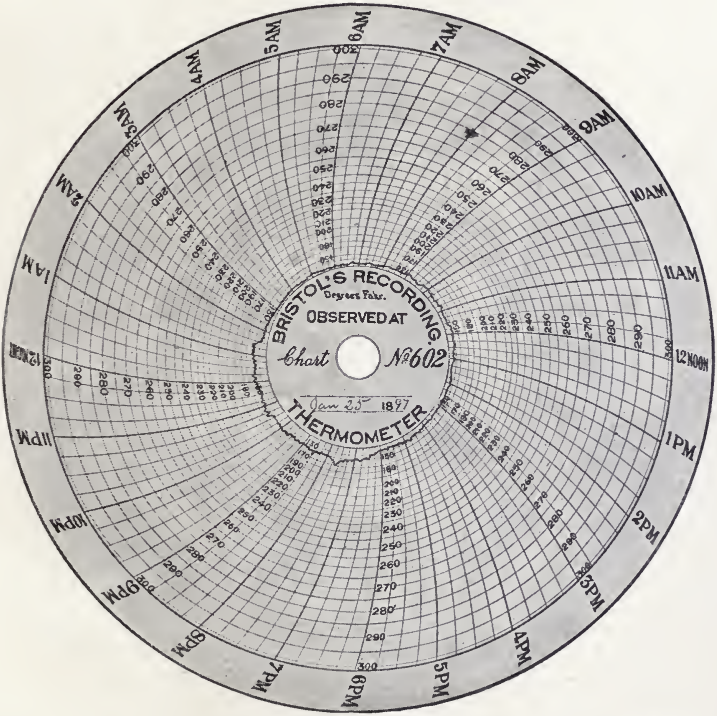

Fig. 74. Air-Temperature Chart from Air Heater at Union Traction Company, Philadelphia, Pa.

direct passages for air and gases are secured by placing the induced-draft fan upon a platform above the heating fan. The hot air, leaving the end of the heater, is conducted through galvanized iron pipes to various parts of the buildings, and there properly distributed. A suitable arrangement of dampers makes it possible to shut any desired connection at will. The one which is located near the inlet of the induced-draft fan is operated by an automatic damper regulator.

The coal used is buckwheat, of which from 6 to 8 tons are used every 24 hours. The actual results secured as regards regulation of steam pressure and temperature to which air is heated by passing through the air heater, are very clearly shown in Figs. 73 and 74, which are reproduced directly from charts taken respectively from the recording pressure gauge and the recording thermometer under ordinary working conditions. The former serves to show that the maximum variation in steam pressure does not exceed two pounds. The latter demonstrates the relative uniformity maintained in temperature of flue gases, of which the temperature of the heated air is to a certain extent an index, although influenced not only by the temperature, but also by the velocity of the flue gases passing through the air heater.

Dust Destructors at Shoreditch, London, Eng. — The complete and satisfactory destruction of dust, garbage and similiar refuse material is rapidly becoming one of the most important elements in modern sanitary engineering. In a large number of the successful furnaces constructed for this purpose, it has been found necessary to secure the required intensity of draft and temperature by forcing the combustion by artificial means. For this purpose the Sturtevant fan has proved itself readily adaptable.

[1] "The vestry and parish of St. Leonard's, Shoreditch, has proved itself to be extremely enterprising, and is building upon a very central site a combination of municipal undertakings. Of these perhaps the most important is the electricity and dust-destruction undertaking, which was formally opened on Monday, the 28th June, by Lord Kelvin, in the presence of a numerous gathering."

After finally acquiring the site, "the lighting committee forthwith instructed Messrs. Manlove, Olliott & Co., Ltd., to report as to what could be done with the type of dust destructor manufactured by them, and what results they could guarantee to obtain by burning 20,000 tons of refuse per annum. The firm reported that with the thermal storage system of Mr. Druitt Halpin such refuse could produce sufficient heat for the electric lighting station proposed by Mr. Manville, and the value of the steam so produced would be £4,290 per annum, and that a saving of at least £1,500 per annum could be effected by burning refuse instead of barging it away, as was then being done. . . . Formerly they had to pay 3*s.* 2*d.* per ton of refuse for barging it away, but now it would only cost 1*s.* 2*d.* per ton for burning it in the dust destructor, — an obvious saving of 2*s.* per ton."

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[1] Electricity and Dust Destruction in Shoreditch. The Engineer, London, July 2, 1897.

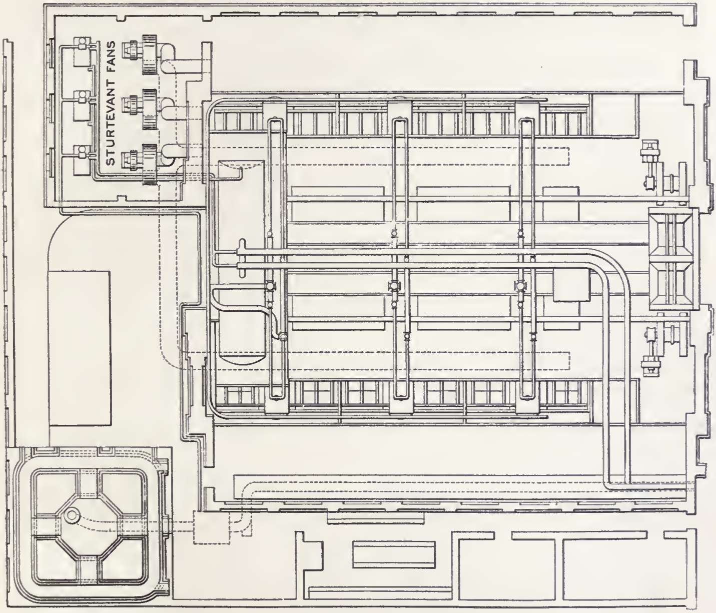

Fig. 75. Plan of Destructor House, Showing Sturtevant Fans at Shoreditch, London, Eng.

"The destructor house [see Fig. 75] is 80 feet square, and contains 12 cells, each having 25 square feet grate area, and 6 water-tube boilers, each with 1,300 square feet of heating surface. The boilers and thermal storage vessel — which is 35 feet long and 8 feet in diameter — are designed to work at a pressure of 200 pounds per square inch, and are supplied with duplicate fittings throughout to guard against breakdown. There are 3 [Sturtevant] motor-driven fans calculated to deliver each 8,000 cubic feet of air per minute with a maximum ashpit pressure of 3 inches of water."

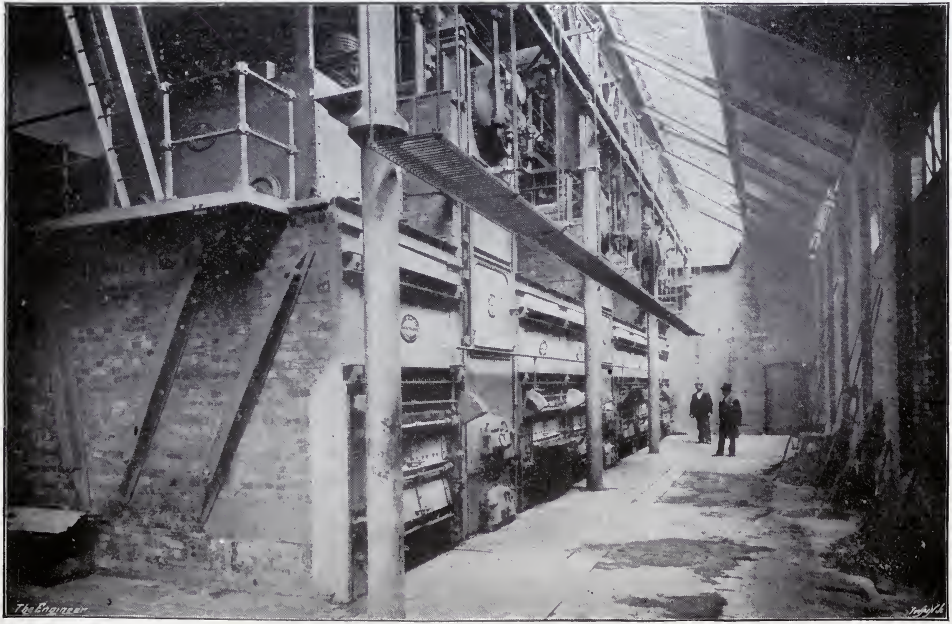
The Engineer

FIG. 76. DUST DESTRUCTORS AT SHOREDITCH, LONDON, ENG.

The general arrangement of the three Sturtevant fans, which per the above are intended to work at high pressure, is clearly shown in Fig. 75. They are all of the downward-discharge type, delivering the air to underground ducts extending beneath them, and hence distributing it to the destructor cells. The inlets of the fans are arranged to draw their air supply from the main sewer, thereby ventilating the same for a considerable distance in the neighborhood of the destructor works. Connection is also made to the inlet trunks from the space immediately above the cells, which in hot weather becomes almost unbearable for the men employed in dumping the rubbish, unless the fans are used to draw the hot air away. In Fig. 76 (from "The Engineer") is shown the front of the battery of destructor cells and boilers; the fans being located in the room to which the door at the farther end of the boiler-room admits.

The economy and utility of forced draft are shown in this connection by the fact that the fans were proportioned on the estimated requirement of only 190 cubic feet of air per pound of refuse, and a rate of combustion of 26 pounds of refuse per square foot of grate per hour.

United States Cotton Company, Central Falls, R. I. — This is the plant already referred to in Chapter V. as showing a weekly saving of over $126.00 resulting from the introduction of mechanical draft. It was there shown that this result was principally due to the burning of cheap fuel, which was only made possible by the introduction of the fan. There is here presented the record of two consecutive tests of 13 weeks' duration each, showing economic results that are worthy of the closest attention, and agreeing with remarkable closeness as to the exact average cost per indicated horse-power for the entire period.

[1] "This boiler plant consists of a battery of three Babcock & Wilcox water-tube boilers, each 18 tubes wide and 9 tubes high, with three steam drums 36 inches in diameter. The furnaces are 10 feet 10 inches by 7 feet. These boilers are equipped with automatically controlled mechanical draft, air being supplied to the ashpits through a blast box 60 inches by 5 inches in the bridge wall of each boiler, by means of a special Sturtevant 90-inch fan, driven by a direct-connected cross-compound 4–7 x 4 upright enclosed engine. The steam pressure carried on this plant is 150 pounds. The engine is a cross-compound condensing of the Harris-Corliss type.

"During thirteen weeks, ending April 3, 1897, the engine ran up to speed 773.94 hours, developing an average of 1,562.36 indicated horse-power, as shown by cards taken twice per day. All the fuel fired to the furnaces, for

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[1] Thomas P. Burke, chief engineer of United States Cotton Company, Central Falls, R. I. Letter of Oct. 6, 1897, to B. F. Sturtevant Co.

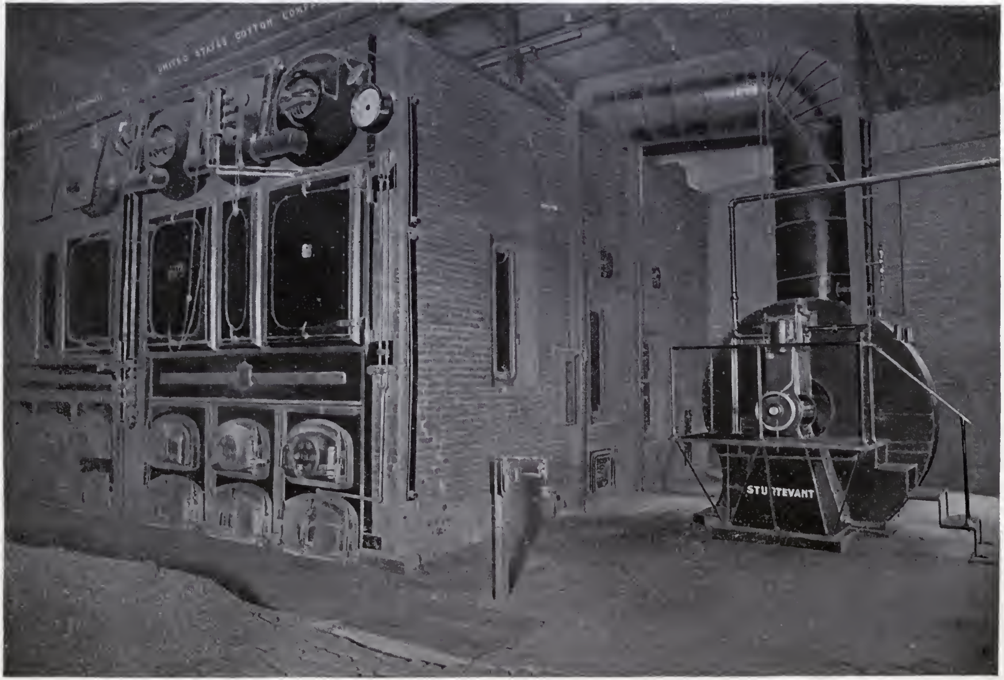

UNITED STATES COTTON COMP
STURTEVANT

FIG. 77. ARRANGEMENT OF STURTEVANT STEAM FAN FOR FORCED DRAFT AT UNITED STATES COTTON COMPANY, CENTRAL FALLS, R. I.

banking fires at night and getting up steam in the morning, is charged to the time and horse-power of the main engine, as follows: —

	Weight in Pounds.	Cost.
Dry buckwheat coal, containing an average of 17% of dust or anthracite culm that would pass through a $\frac{3}{16}$-inch square-mesh screen	1,613,127	$2,019.91
Dry dust or anthracite culm	287,781	215.48
Low-grade Cumberland	170,983	313.11
Total	2,071,891	$2,548.50

Making a cost of $\frac{210}{1,000}$ of a cent per indicated horse-power per hour. The ashes and unconsumed coal taken from the furnaces and ashpits at night averaged 12.86 per cent.

For the thirteen weeks ending Oct. 2, 1897, the conditions were substantially the same. The engine ran to speed 708.94 hours, developing an average of 1,545.24 indicated horse-power. The fuel fired to the furnaces was —

	Weight in Pounds.	Cost.
Dry buckwheat coal containing 18% of dust or anthracite culm	1,518,549	$1,927.50
Dry dust or anthracite culm	253,253	190.90
Low-grade Cumberland	94,934	142.51
Total	1,866,736	$2,260.91

Making a cost of $\frac{206}{1,000}$ of a cent per indicated horse-power per hour. The ashes and unconsumed coal taken from the furnaces and ashpits was 12.9 per cent.

"The temperature of the flue gases directly after leaving the boiler is from 380° Fahr. to 415° Fahr. The blast pressure in the air duct is from zero, when the fan is running very slowly, to 1.3 inches of water when it is speeded up. The average evaporation from and at 212° per pound of combustible was 10.28 pounds, metered through a Worthington hot-water meter, calibrated, and allowance made for slip. During part of the thirteen weeks, ending April 3, the meter was out of commission for repairs, so no evaporation is given. By the term 'dry coal' is meant coal atmospherically dry; that is, dried in the open air."

The general results of these continuous tests are presented in Table No. 132. From the hourly cost per indicated horse-power there given it is evident that the fuel cost per year per indicated horse-power (based on 58 working hours per

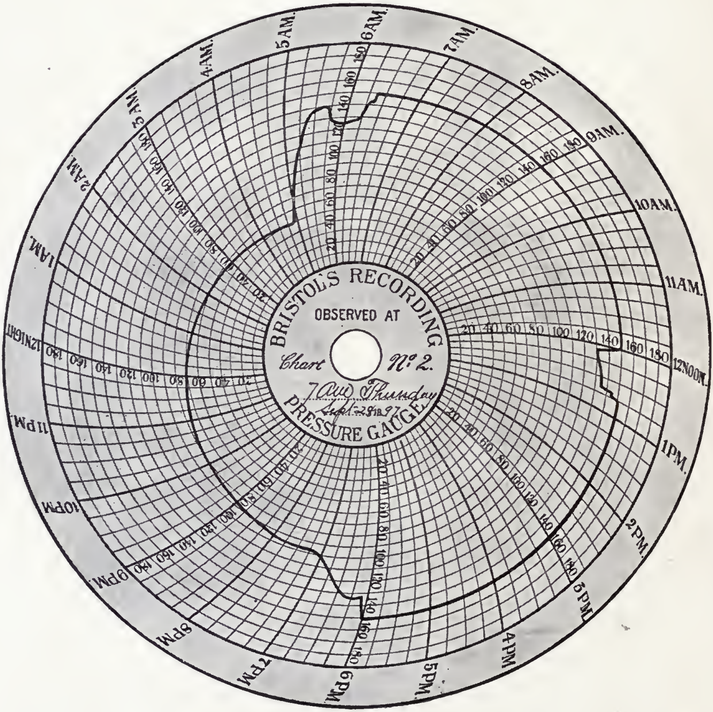


FIG. 78. STEAM-PRESSURE CHART FROM FORCED-DRAFT PLANT AT UNITED STATES COTTON COMPANY, CENTRAL FALLS, R. I.

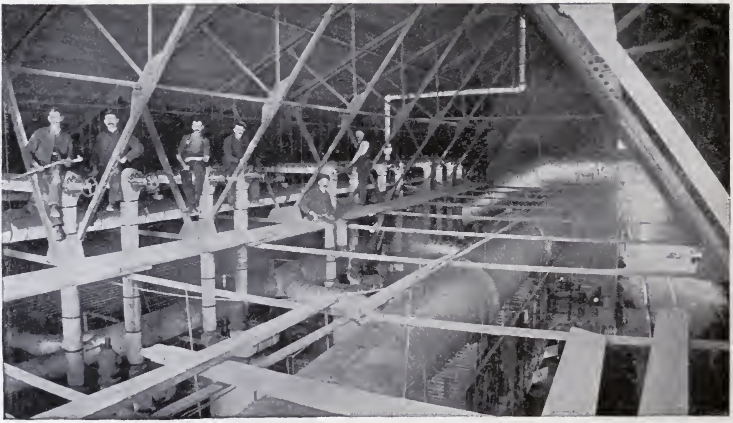
week and 52 weeks per year) is respectively only $6.33 and $6.21, as indicated by the results of the two tests.

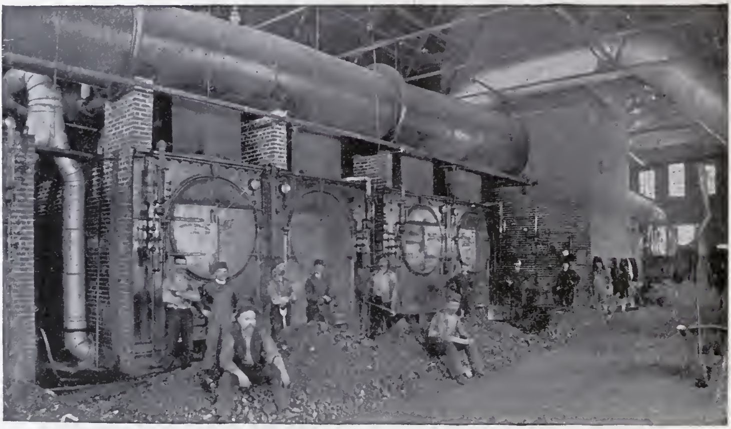
Table No. 132. — **Results of Two Thirteen-Week Tests of Steam Plant with Sturtevant Mechanical Draft, at United States Cotton Company, Central Falls, R. I.**

ITEMS.		Test No. 1.	Test No. 2.
Duration	hours,	773.94	708.94
Average steam pressure	pounds,	150	150
Average I. H. P. of compound engine		1,562.36	1,545.24
Total fuel fired	pounds,	1,613,127	1,518,549
Ash and unconsumed coal	per cent,	12.86	12.9
Water evaporated from and at 212° per pound of combustible,	pounds,		10.28
Total cost of fuel		$2,548.50	$2,260.91
Cost of fuel per hour for one I. H. P.		$0.0021	$0.00206

The front of a portion of the battery of boilers, together with the Sturtevant fan, is clearly shown in Fig. 77. The fan is of the angular down-blast pattern, the air being discharged into an underground duct extending beneath the boilers. The air supply for the fan is drawn from above the boilers through the pipe shown in the illustration. This serves to keep the upper part of the boiler house cool and well ventilated. The perfection of steam-pressure regulation secured by the use of the fan is most emphatically shown by the steam-pressure record, reproduced in Fig. 78. This result is largely due to the method of regulation of the speed of this engine. This is accomplished by means of a device especially designed and patented by the engineer, Mr. Thomas P. Burke, which instantly changes the speed of the fan to correspond with very slight variations in the steam pressure.

Glens Falls Paper Mill Company, Fort Edward, N. Y. — The plant shown in Figs. 79 and 80 serves as an excellent illustration of the manner in which under-grate forced draft can be applied in an existing plant. The boilers are eleven in number, 72 inches in diameter by 19 feet long over all, arranged in five batteries, four of these having two boilers each and one having three boilers. A special 7 x 4 Sturtevant steel-plate steam fan, located in a room opposite the centre of the row of boilers, delivers the air into a pipe which passes upward at an angle, and thence over to a point above the boilers, where it divides and extends the entire length of the plant. From this pipe branches are carried down between the groups of boilers and enter the ashpits at the sides, each pipe being provided with a blast grate.





Figs. 79 and 80. Forced-Draft Plant at Glens Falls Paper Mill, Fort Edward, N. Y.

Crane & Breed Manufacturing Company, Cincinnati, Ohio. — One type of under-feed mechanical stoker used in connection with a Sturtevant fan has already been described. In Fig. 81 is shown a plant equipped with the American Stokers. A No. 5 Sturtevant "Monogram" blower, driven by a 4 x 4 Sturtevant upright engine, serves to supply the requisite air under the pressure which is necessary with this type of stoker, the general form and construction of which is shown in Fig. 82. [1] "Immediately beneath the coal hopper, and communicating with it, is the conveyor, this in turn communicating with the magazine in direct line with it. A screw conveyor or worm is located in the conveyor chamber, and extends nearly the entire length of the magazine. Immediately beneath the conveyor chamber is located the wind-box, having an opening beneath the hopper. At this point is connected the piping for air blast. The other end of the wind-box

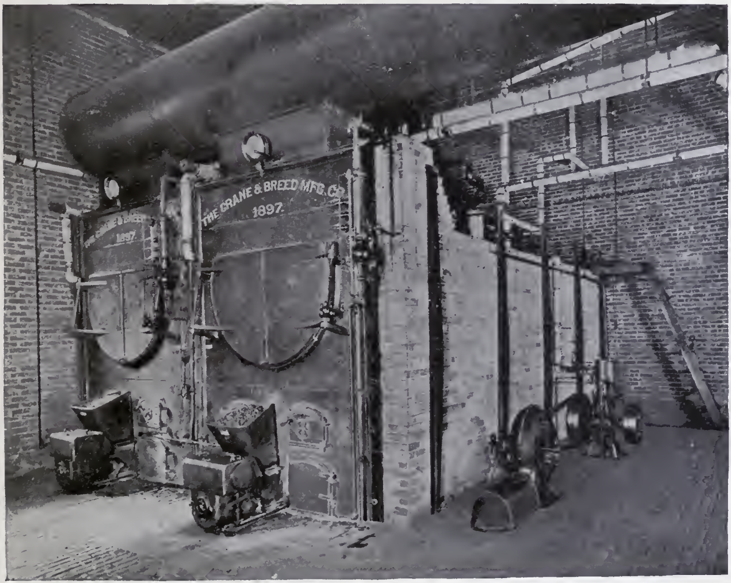


Fig. 81. Arrangement of American Stokers and Sturtevant Fan at Crane & Breed Manufacturing Company, Cincinnati, Ohio.

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[1] The American Stoker. Catalogue, 16 pp. American Stoker Co., Dayton, Ohio.

opens into the air space between the magazine and outer casing or envelope. The upper edge of this magazine is surrounded by tuyeres or air blocks, these being provided with openings for the discharge of the air blast.

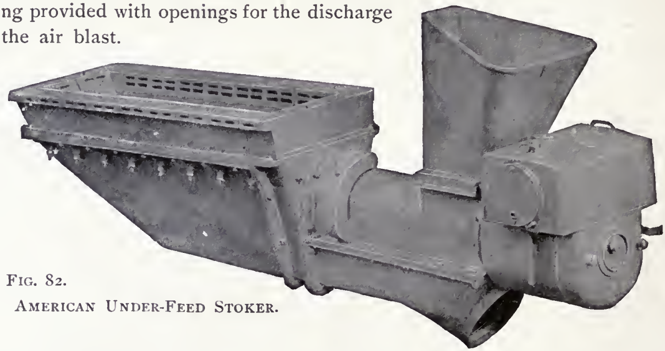

FIG. 82.
AMERICAN UNDER-FEED STOKER.

"The space on each side of the stoker, between the tuyere blocks and the side walls of the furnace, is occupied by dead plates or air-tight grates. The coal is fed into the hopper, carried by the conveyor into the magazine, and is there forced upward, 'overflowing' on both sides, and spreading upon the dead grates the entire width of the furnace [as shown in Fig. 83]. The entire mass of coal above the tuyeres and all of that upon the dead grates is ignited, carrying a bed of burning coke from 14 to 18 inches in depth.

"We use the volume blower for air, and actuate this either by a small engine or a convenient line shaft. The air is delivered in the approximate proportions of 150 cubic feet of air to each pound of coal fed, and at a pressure ranging from ½ ounce to 1 ounce at the tuyeres. This pressure is only such as to admit of the thorough mixing of the air with the coal, and is in no sense of the word a blast. A wind-gate, controlled by a lever, enables the operator to regulate the supply of air to suit the amount of coal fed. Being thus independent of

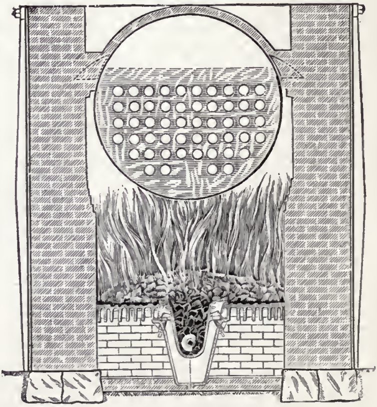

FIG. 83. CROSS SECTION, SHOWING FURNACE IN OPERATION.

natural draft, and the supply of coal under complete control, the fire can be forced at a moment's notice, and can be as quickly reduced.

[1] "As the result of a long series of chemical analyses of the chimney gases, collected under precisely similar conditions, both when fired by hand and by stoker, it is shown that the amount of air required per pound of coal when burned with this stoker, is from 20 to 55 per cent less than that which would have been used in the common hand-fired practice. This effects a two-fold economy in decreasing the volume of heated gases passing up the chimney and likewise decreasing the velocity of the gases as they pass over the heating surfaces, thus allowing more time for the absorption of heat by the boiler surfaces. Therefore, the increase of economy must somewhat exceed the product resulting from multiplying the actual percentage of decrease of air by the corresponding chimney losses.

"This economic use of air is due to the method of operation peculiar to this stoker, in that it carries a bed of coal of unusual depth, the air being supplied from underneath, the volume of which being under perfect control, and the continuous feeding action completely overcoming the natural tendency of holes burning through the fire."

Holyoke Street Railway Company, Holyoke, Mass. — The general arrangement of this plant is very clearly shown in Fig. 84. The boilers are at present three in number, of the Babcock & Wilcox water tube type, of 200 horse-power nominal rating each, set in a single battery, and operating under 125 pounds steam pressure. Eventually another battery of three will be in a corresponding position on the other side of the fans. The gases, after leaving the boilers, pass through a Green economizer, and thence to the brick chamber beneath the two Sturtevant steam fans. A damper in the connection between the fans, and another in the outlet connection above, make it possible to pass the gases through either or both of the fans. By a special arrangement of automatic control, the speed is so regulated as to maintain a practically constant steam pressure.

Although the demand upon the fans has as yet been comparatively light, because only half of the proposed boilers have been installed, nevertheless the adaptability of mechanical draft to extremely variable conditions of electric railway work has been emphatically proven. It is stated that they [2] "are having good results as compared with statements from other stations. The forced draft works nicely, and we think that it 'fills the bill' thus far.

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[1] The American Mechanical Stoker. C. H. Bierbaum, The Electrical Engineer. New York, November 18, 1896.

[2] Holyoke Street Railway Company, Holyoke, Mass. Letter of March 26, 1896, to B. F. Sturtevant Co.

In February we ran a daily average of . .	19.14 hours.
Coal consumed per hour	524 pounds.
Electrical horse-power per hour	243 horse-power.
Coal consumed per electrical horse-power per hour,	2.19 pounds.

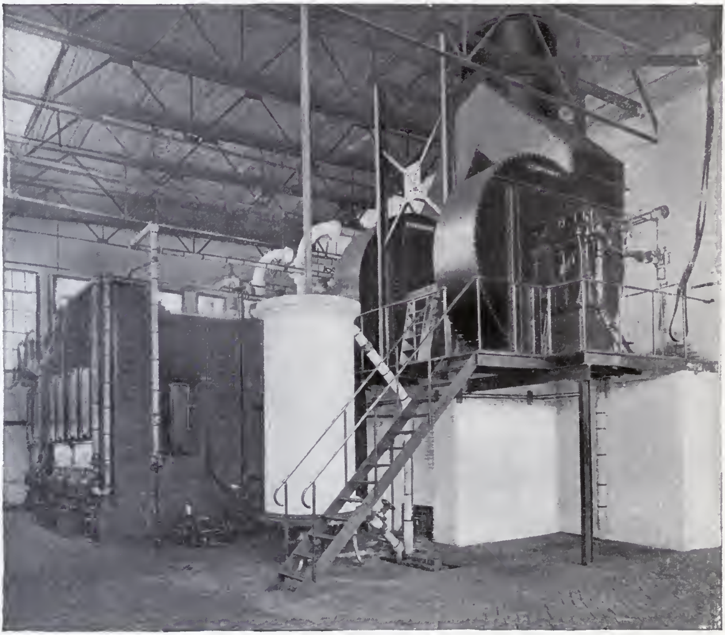

FIG. 84. INDUCED-DRAFT PLANT WITH STURTEVANT FANS AT HOLYOKE STREET RAILWAY COMPANY, HOLYOKE, MASS.

[1] "We know that it has more than fulfilled our expectations, and that it is more economical than represented by you to be. It is extremely convenient to

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[1] Holyoke Street Railway Company, Holyoke, Mass. Letter of August 5, 1897, to B. F. Sturtevant Co.

have precisely the draft desired at any time, and in all kinds of weather. The repairs have amounted to very little indeed, and the whole of it, in a nutshell, is that we are much pleased with it, and would not exchange it for a huge chimney for any consideration."

From the records for the month of August, 1897, the average results in Table No. 133 have been determined. The plant is usually started up about 5.15 a. m. and stopped at 12.15 a. m., and all three boilers are seldom used.

Table No. 133. — Average Results of Record of Steam Plant with Sturtevant Induced Draft at Holyoke Street Railway Company, Holyoke, Mass.

ITEMS.		Average for Month of Aug., 1897.
Total time in operation	hours,	569
Total coal consumed	pounds,	415,591
Total ash	pounds,	30,897
Total combustible consumed	pounds,	384,694
Average coal consumed per hour	pounds,	730.4
Total water pumped into boilers	pounds,	4,150,672.4
Average water pumped into boilers per hour	pounds,	7,294.7
Average water pumped into boilers per pound of coal consumed	pounds,	9.99
Average temperature of water entering economizer	degrees Fahr.,	208
Average temperature of water entering boiler	degrees Fahr.,	242.7
Total electrical output	Watts,	154,150,000
Average electrical output per hour	Watts,	270,900
Equivalent electrical horse-power per hour		363.1
Average coal per electrical horse-power per hour	pounds,	2.12

Farr Alpaca Company, Holyoke, Mass. — This plant, which is located at the No. 1 mill, presents a somewhat novel arrangement of fans, whereby a relay is provided and the floor area occupied is reduced to a minimum. Each fan is of special form, having a wheel 7 feet in diameter, and being driven by a direct-connected 6 x 5 double upright engine. By means of an arrangement of dampers, the gases may be caused to pass through the economizer, and thence to either one or both of the fans, whence they are discharged through the short, vertical stack, or in case the economizer is out of repair, or it is not desired to use it, the gases may be by-passed and enter the fans direct. The general arrangement of the complete plant is illustrated in Fig. 85.

As stated,[1] "it was decided to use mechanical draft on this plant principally because there was very poor foundation for a chimney, and very little room to

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[1] Samuel M. Green, Consulting Engineer. Letter of Oct. 11, 1897, to B. F. Sturtevant Co.

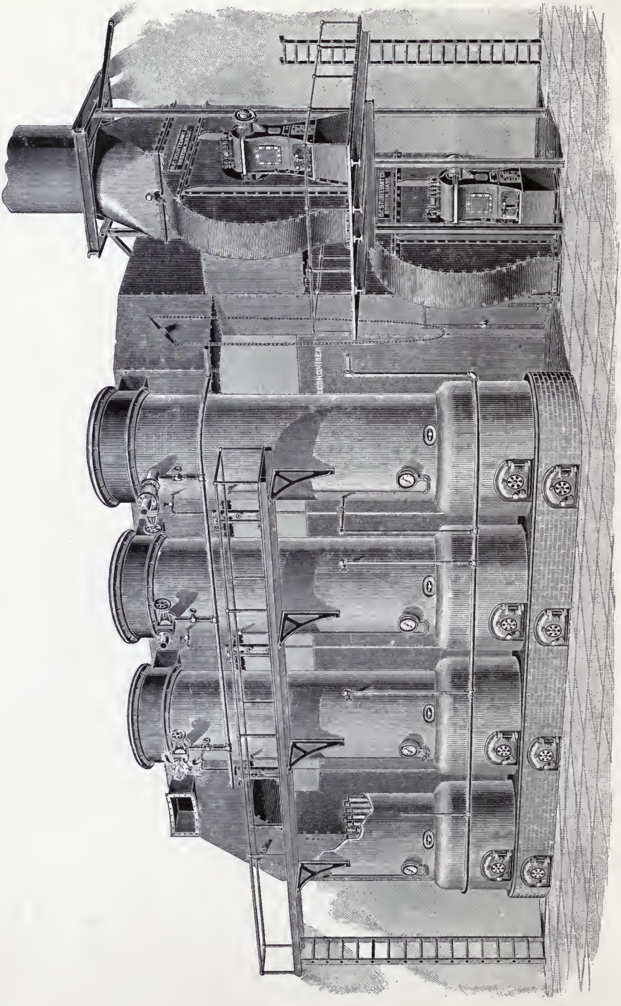

Fig. 85. Induced-Draft Plant with Sturtevant Fans at Farr Alpaca Company, Holyoke, Mass.

place one. The fans were put in of ample capacity to handle four Manning boilers, of which there are two now in position; one more is about to be installed. Since the fans were placed in the boiler room and gotten into working order, they have been very satisfactory. The boilers are of the Manning upright type, each boiler containing 180 2½-inch tubes, 15 feet long; fire box 6 feet in diameter. The area of the grate is 28.27 sq. ft.; the heating surface in each boiler is 1,823 sq. ft. The economizer contains 192 tubes 4⅝-inches diameter, 4 pipes wide, and 48 sections. The square feet of heating surface is 2,304.

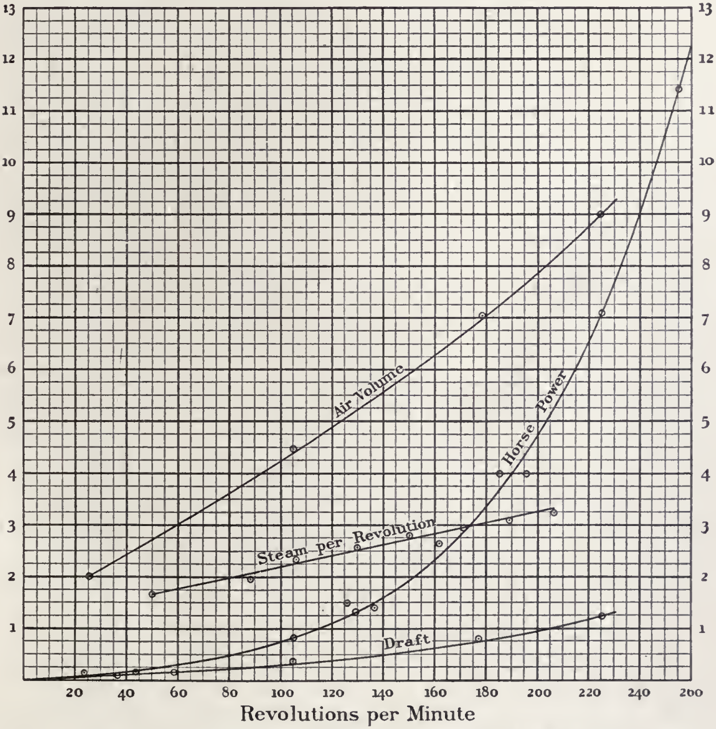

FIG. 86. RESULTS OF TESTS OF INDUCED-DRAFT PLANT AT FARR ALPACA COMPANY, HOLYOKE, MASS.

"I have made a series of tests upon the fan and fan engines to determine the power of the draft, the horse-power consumed by the fan engines and the steam used per horse-power."

The results thus obtained furnish an interesting commentary upon the relations between fan speed, volume moved, pressure created and horse-power required, which relations have already been discussed at length in a preceding chapter. In Fig. 86 these results are graphically presented in such a manner as to make evident the conditions under which the various changes took place. No record of temperatures was taken, and consequently accurate comparison cannot be made. The air volume was measured at the ashpit doors by means of an anemometer. The curve displays the relative volumes admitted, but obviously does not indicate the amount which passed through the fans. The rapid increase in the power required when the speed and resultant pressure increase is clearly shown, both by the curve upon Fig. 86 and even more clearly by the accompanying reproductions in Fig. 87, of cards taken from one of the engines at the progressive speeds given beneath the respective cards. Up to

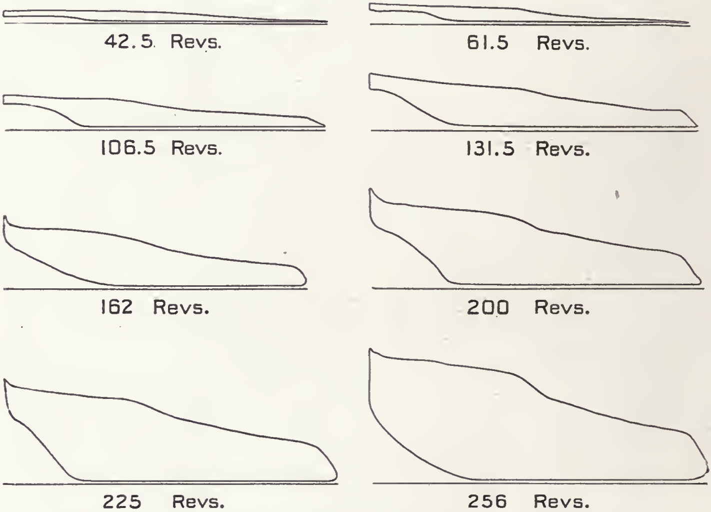
42.5 Revs.
61.5 Revs.
106.5 Revs.
131.5 Revs.
162 Revs.
200 Revs.
225 Revs.
256 Revs.

Fig. 87. Indicator Cards from Sturtevant Engine on Induced-Draft Plant at Farr Alpaca Company, Holyoke, Mass.

a certain speed, the natural draft of the short stack is equal to, or actually exceeds, that created by the operation of the fans; as a consequence, the work done at the lower speeds is very slight indeed. But when the draft produced by the fans exceeds that which the stack is capable of creating, the additional work is thrown upon the fans, and the power increases practically as the cube of the number of revolutions.

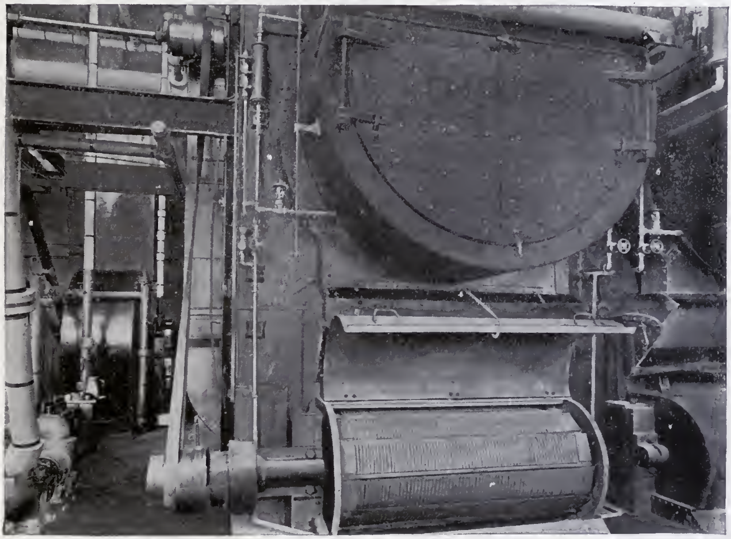

FIG. 88. ARRANGEMENT OF STURTEVANT FAN FOR MECHANICAL DRAFT AT POPE TUBE COMPANY, HARTFORD, CONN.

The Pope Tube Company, Hartford, Conn. — This boiler plant, which is a model in its way, is thus briefly described: [1] "At the present time there are installed four horizontal tubular boilers, 78 inches in diameter by 18 feet in length, each containing 151 3-inch tubes, 18 feet long, and designed to work under a pressure of 135 pounds. The grates are of the Coxe travelling type, 6 feet wide, and move at a maximum rate of 12 feet per hour. The coal used is No. 2 buckwheat anthracite.

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[1] The Iron Age, New York, March 4, 1897.

"The draft is forced by a Sturtevant 66-inch fan direct-connected to a single-cylinder 5½ x 8¼-inch engine, which also drives the grates. The speed of the fan is regulated automatically by the steam pressure acting through a Locke damper regulator, so that the rate of grate travel and blast pressure keeps the steam pressure constant, irrespective of the demand for steam."

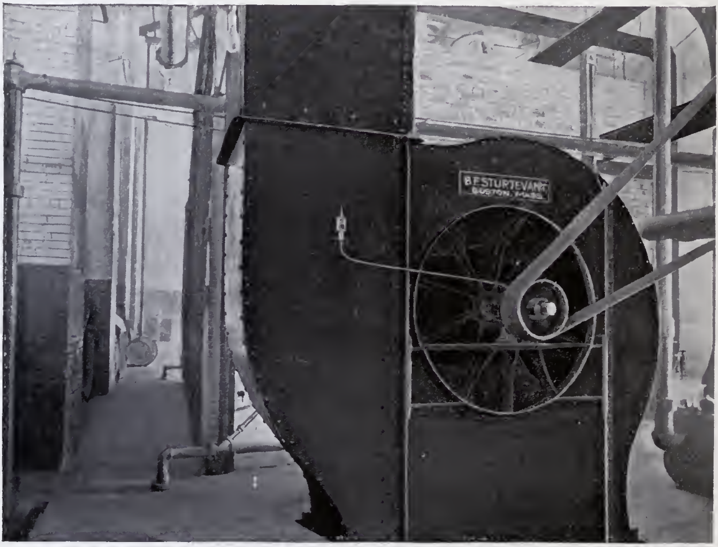

FIG. 89. STURTEVANT FAN FOR MECHANICAL DRAFT AT POPE TUBE COMPANY, HARTFORD, CONN.

In Fig. 88 is presented a front view of one of the boilers, to the left of which and in the rear is shown the Sturtevant fan with direct-connected engine. A full view of the inlet side of this fan is shown in Fig. 89. The air is discharged upward from the outlet of the fan, enters a horizontal main above the boilers and thence is delivered to individual rectangular pipes which connect with the ashpits. One of these pipes is shown in each of the illustrations. The chambered arrangement of the ashpit, within which various pressures are maintained, has already been described in connection with the report upon the Deringer Colliery. A pulley upon the extended shaft [see Fig. 89] serves as a means of driving the travelling grate.

S. S. St. Louis and St. Paul. — These twin ships of the American Line, International Navigation Company, are each equipped with eight special Sturtevant steel-plate steam fans for the production of the requisite draft. The ships are twin-screwed, each being provided with 2 six-cylinder quadruple expansion engines, having cylinders as follows: two high pressure, 28½ inches diameter, one first intermediate, 55 inches diameter, one second intermediate, 77 inches diameter, and two low pressure, 77 inches in diameter. [1] "Steam for the main engines is supplied by ten boilers of the Scotch type, six of which are double-ended and four single-ended. They are all about 15½ feet in diameter; the double-enders being 20 feet long, and the single about half that length. Each boiler has four furnaces, eight of course in the double-enders, making 64 furnaces in all, each with 18 square feet of grate, giving a total grate surface of 1,144 square feet. The total heating surface is 40,320 square feet, giving a ratio of a little over 36. Imagine a surface 200 feet square covered with boiling water with a fire 35 square feet below it forced to a white heat by a hot blast, and burning 300 tons of coal a day, and you have an idea of the magnitude of the steam-generating plant of one of these magnificent vessels. The boilers are arranged in two groups, or batteries, each battery in a water-tight compartment. They set fore and aft, or lengthwise of the ship, three of the boilers side by side and two of the small ones facing them, in each compartment. . . . The Howden system of forced draft is used. Each stokehole, of which there are four, is supplied with two 80-inch Sturtevant fans, each driven by two-cylinder 8 x 5½-inch engines directly connected to the fan shaft. [See Fig. 91.] These fans draw the air from the top of the stokeholes, force it through the tubes in the uptakes of the boilers, and pass it to the casing or chamber, which will be seen protruding from the boiler front."

The general arrangement of these tubes in the uptakes is very clearly shown in the accompanying Fig. 90, which is from a photograph of one of the double-ended boilers for the St. Paul. The fans are of special construction and rigidly attached to the water-tight bulkheads.

"Directly above each furnace door, a small crank will be seen. This controls the admission of air to a chamber in the door of the furnace, whence it issues through the perforated plate and is delivered over the fire. At the side of each furnace is another handle which controls the admission of air below the grate. . . . The fire tubes are provided with spiral deflectors [retarders] to retard the passage of gases and keep them impinging against the heating surface."

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[1] Steam Plant of the St. Paul and St. Louis. Power, New York and Chicago. February, 1896.

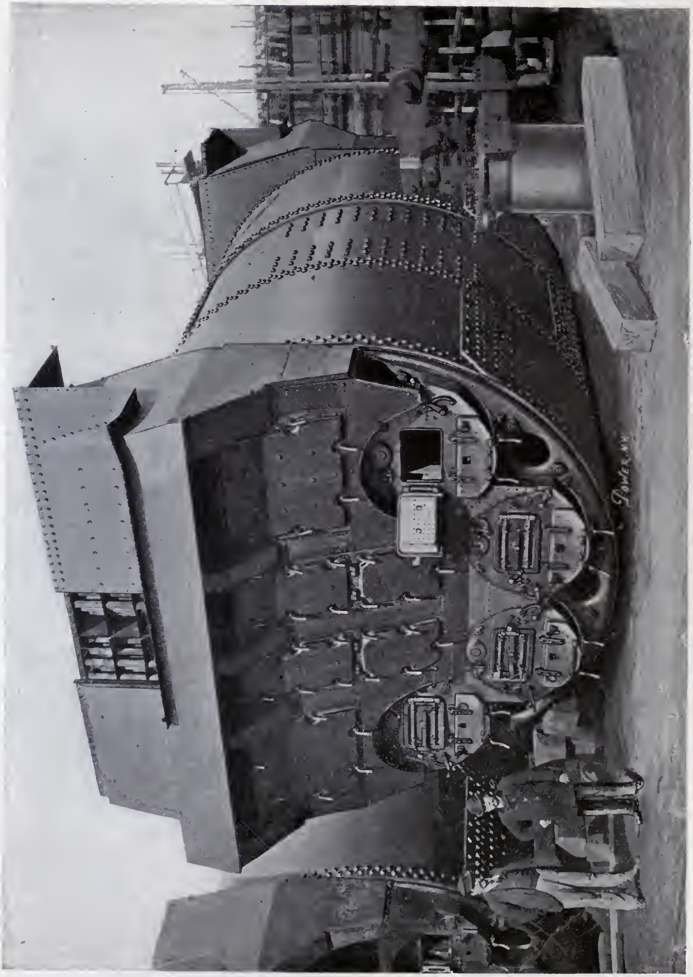

Fig. 90. Boiler Equipped with Howden Hot-Draft for use with Sturtevant Fans on S. S. St. Paul of American Line, International Navigation Co.

The chief advantages claimed for this system are [1] "(1) increase of power, (2) economy in fuel, (3) reduced wear and tear of boilers, (4) coolness in fireholds, (5) reduced size and weight of boilers for a given power, (6) simplicity.

"First. The power from a boiler of a given size may be increased by the use of this system with safety and comparative economy 40 per cent over that obtainable by natural draft. Mr. Howden claims in special cases 100 per cent increase, but it is, however, only in special cases that such a great increase is necessary or desirable. In ordinary merchant steamers from 40 to 60 per cent increase of power is obtained, or from 16 to 19 indicated horse-power per square foot of fire grate. Such powers are obtained with a very high economy in fuel.

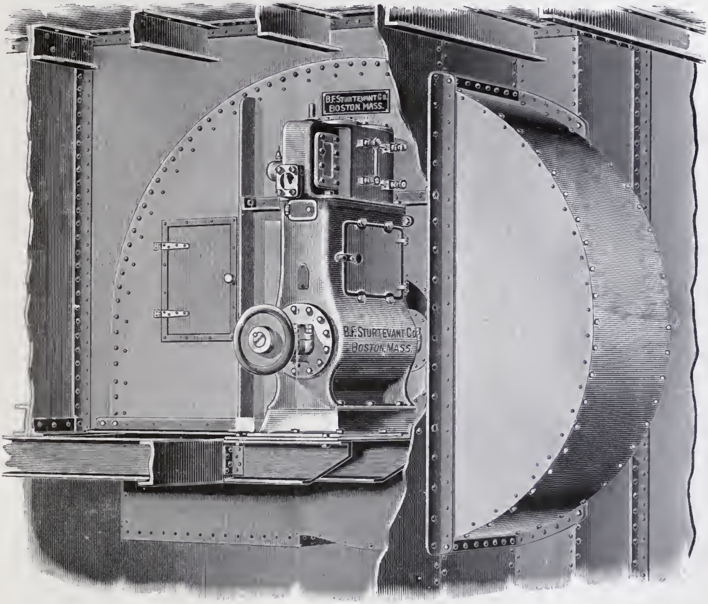
B.F. STURTEVANT Co.
BOSTON MASS.
B.F. STURTEVANT Co.
BOSTON, MASS.

FIG. 91. STURTEVANT SPECIAL STEEL-PLATE STEAM FAN FOR FORCED DRAFT ON S. S. ST. LOUIS AND ST. PAUL.

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[1] The Howden System of Hot Draft. Dry Dock Engine Works, Detroit, Mich. November, 1896.

"Second. The high economy in fuel obtained by this system of combustion is one of its most valuable features, and has contributed largely to its adoption.

"The greatly superior economy of the Howden draft system, compared with the best examples of natural draft, has been completely proved by the British India Steam Navigation Company, who built two large steamers for the purpose of comparing results. These steamers had hulls and engines exactly alike, the only difference being in the boilers: one of the steamers, Vadala, having two double-ended boilers with eight furnaces worked by natural draft; the other vessel, the Virawa, having two single-ended boilers with four furnaces worked by the Howden system.

"The steamers were worked in the same trade, and all particulars of consumption, speed, etc., carefully noted by the company over a period of three years. Every precaution was taken to arrive at a correct comparison by changing engineers from one ship to the other, etc.

"The result of over three years' working was found to be that, while the Virawa, with the Howden hot draft, had a fully higher average speed, her consumption of fuel was fully 20 per cent less than the Vadala with natural draft boilers. The effect, however, of the improvements on our system, which we have introduced during the last few years, has been to greatly increase this economy. The economy of the Howden system is supposed by many engineers to be measured merely by the units of heat utilized by heating the air of combustion from the waste gases. This view entirely overlooks the value of several important collateral effects which contribute largely to the economy of the system. These will be understood better by the following explanation:—

"By whatever amount the air of combustion is increased in temperature by the waste gases, the average temperature of the furnaces is practically raised to the same extent.

"From this increased furnace temperature several distinct economical effects arise: (1) By increasing the evaporative efficiency of the heating surface. . . . (2) The gases from the burning fuel combine more readily with the oxygen of the air of combustion as the temperature of the fire increases, and consequently less air is required for combustion per unit of coal at the higher temperature. (3) This reduction in quantity of air required per unit of coal consumed has also an important economical effect. The furnace temperature is increased by having less air to heat to the furnace temperature; and less heat is likewise carried off by the chimney gases. Further, the volume of gases passing through the boiler being less in a given time, its velocity is less, and thus the hot gases are longer in contact with the evaporating surface, and impart a greater proportion of heat to the water.

"Third. The reduction in wear and tear of boilers arising from the use of our system is now also well established. . . . The following facts confirm the durability of boilers worked with the Howden system of draft: The first boiler (marine type), made thirteen years ago for experimental purposes, has since supplied steam continuously for driving the Howden works. . . . The first boiler fitted in a steamship with our system was that of the New York City. This steamer was sold not long since with the boiler in excellent order, with the original tubes, furnaces and combustion chambers.

Table No. 134. — Results from Log of Round-Trip Voyage of S. S. St. Louis, Operating under Forced Draft with Howden System and Sturtevant Fans.

ITEMS.	Voyage No. 33.	Voyage No. 34.
Direction bound	East.	West.
Date of departure	Sept. 1, 1897.	Sept. 11, 1897.
Date of arrival	Sept. 8, 1897.	Sept. 17, 1897.
Time of passage, dock to dock . . days, hrs., mins.,	6:14:29	6:11:15
Time of passage, sea . . days, hrs., mins.,	6:10:14	6:7:1
Average steam pressure . . pounds,	198	198
Average pressure, fan discharge . . inches of water,	2.5	2.5
Average pressure in air reservoirs . inches of water,	2.0	2.0
Average pressure in ashpits	1.5	1.5
Average indicated horse-power	17,863	20,768
Knots per hour	19.95	20.22
Coal per I. H. P. . . pounds,	1.59	1.59
Coal per hour per square foot of grate . . pounds,	24.88	28.88
Indicated horse-power per square foot of grate	15.61	18.15
Indicated horse-power per square foot of heating surface	0.443	0.515
Temperature in air reservoirs . degrees Fahr.,	263	263
Temperature in funnel . degrees Fahr.,	564	576
Temperature of atmosphere . degrees Fahr.,	61	61
Temperature of fire room . degrees Fahr.,	105	112

"Fourth. The greater coolness of the fire-holds in steamers using our hot draft is caused by the absence of radiation of the heat of the furnaces and ashpits into the fire-holds.

"Fifth. That the space in steamships occupied by the boilers worked with our system is *very much less* than in steamers having natural draft boilers of equal power is evident from the much greater power obtained with our system from boilers of a given size. The weight of the boilers is proportionately less with our system.

"Sixth. For simplicity this system above all others recommends itself. It is a usual thing, when fitting our system in steamers where two double-ended boilers with two fire-holds have been used for natural draft working, to use two single-ended boilers, with half the number of furnaces, and one fire-hold only, thus saving a large space in the ship, and also reducing the number of firemen and trimmers to about one-half for the same power."

The general results of the operation of this system of mechanical draft are exemplified in Table No. 134, which is compiled from the logs of a round-trip voyage of the S. S. St. Louis. The coal per hour per indicated horse-power as therein given, includes that required for the production of all steam used at sea for the auxiliary engines and other purposes.

S. S. Kensington and Southwark. — These two vessels of the Red Star Line, International Navigation Company, are practically identical in dimensions and design, and are each equipped with five special Sturtevant fans, in connection with the "Ellis & Eaves" system of heat abstraction and induced draft. The general features of this system have already been presented at considerable length, in connection with the description of the plant at the American Line, Pier 14. A description of one of these vessels and its equipment is practically a description of the other, therefore, the following is applicable to both: —

[1] "The S. S. Kensington of the International Navigation Company is a twin-screw vessel of 8,670 tons, and her principal dimensions are — length, 494 feet by 57 feet beam. She has two sets of quadruple expansion engines, the diameter of the cylinders being 25½ inches, 37½ inches, 52½ inches, and 74 inches by 54-inch stroke. The working pressure is 200 pounds per square inch. The steam is generated in three Scotch boilers, two of which are double-ended and one single-ended. The double-ended are 15 feet 9 inches diameter by 21 feet 5 inches long, while the single-ended is of the same diameter but 11 feet 3 inches long. There are four furnaces at each end, making 20 in all, each with a Purvis flue. These are 3 feet 4 inches mean diameter. The length of the grate is 5 feet 9 inches, and the total grate area 383 square feet. The total heating surface is 12,176 square feet. The boilers are fitted with Serve tubes 3¼ inches in diameter, with 1⅜-inch space between. The 'Ellis & Eaves' system of induced draft is fitted, and the boilers are in one compartment and exhaust into one chimney stack, which is 84 feet high from the grate level and elliptical in plan, 14 feet by 9 feet. The fans, of which there are five, are situated at the base of the funnel and are driven direct by Sturtevant engines. They are 7 feet 6 inches in diameter. These fans induce a draft through the

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[1] Chief Engineer Snowdon, of S. S. Kensington. Letter to B. F. Sturtevant Company.

furnaces, the air having previously been heated by passing through tubes placed in a casing over the boilers, in the way of the waste gases from the furnaces. The waste gases, after leaving the furnace, play around these tubes forming the air inlet, and subsequently pass through the fans into the funnel. Each furnace has valves above and below the grate for regulating the supply of *hot air*, and the furnace doors have dampers connected to them so that upon opening the door the draft to that fire is minimized. The fans are so arranged that one may draw from eight fires if necessary. At the trials on the measured mile, the mean draught was 21 feet 8 inches." The general arrangement of the main boilers, fans and heat abstractors is very clearly indicated in the accompanying cross section, Fig. 92.

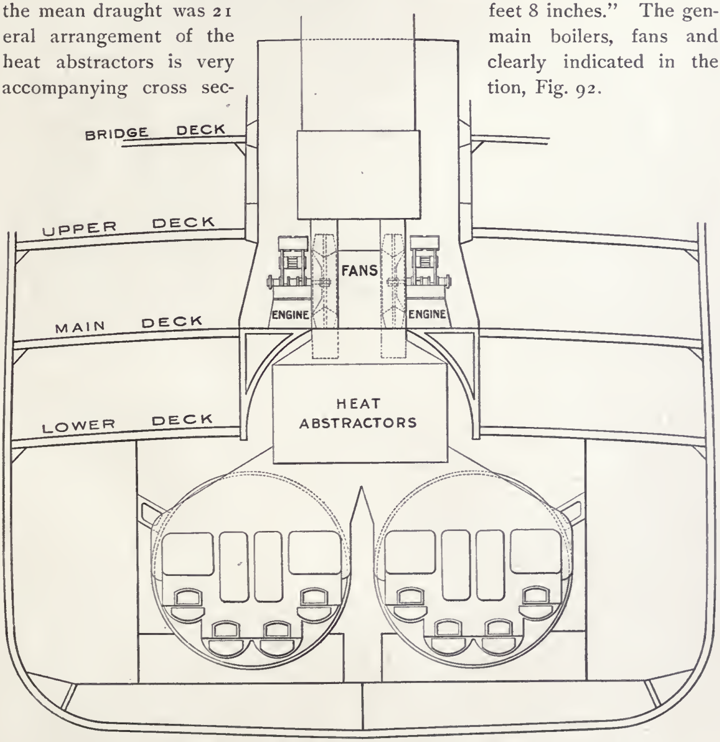

Fig. 92. Arrangement of Main Boilers, Fans and Heat Abstractors on S. S. Kensington.

Four fans are grouped beneath the funnel, as shown in Fig. 93, each fan being continuously driven at high speed by a double-cylindered Sturtevant engine. The hot gases from the abstractors pass to the inlet connections between the fans, and thence through the fans to the funnel above. A fifth fan of the same design is applied to the donkey boiler.

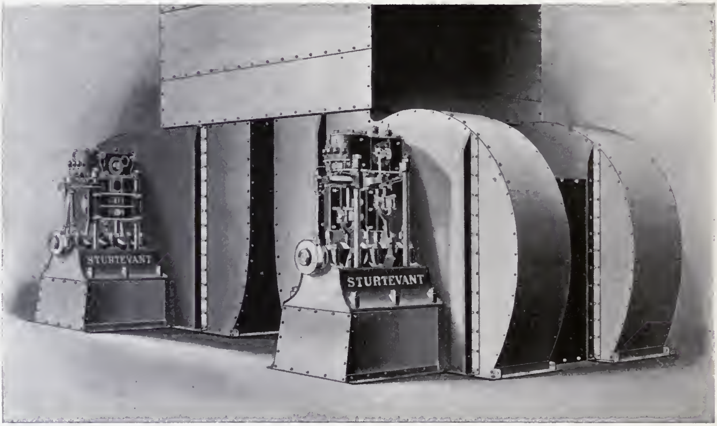

FIG. 93. STURTEVANT STEAM FANS FOR PRODUCTION OF INDUCED DRAFT IN CONNECTION WITH ELLIS & EAVES SYSTEM ON S. S. KENSINGTON.

Table No. 135. — Sample Set of Data for One Day on S. S. Kensington, under Ordinary Working Conditions of Boilers, Using Ellis & Eaves Induced Draft with Sturtevant Fans.

Velocity of air entering furnace	feet per minute,	3,716
Velocity of air entering heating tubes	feet per minute,	895
Temperature of air entering heating tubes	degrees Fahr.,	140
Temperature of air entering fires after passing through heating tubes,	degrees Fahr.,	295
Temperature of waste gases	degrees Fahr.,	446
Vacuum at fan	inches of water,	3.5
Vacuum below grate	inches of water,	0.687
Vacuum above grate	inches of water,	0.75
Average revolutions of Sturtevant fans	revolutions per minute,	391
Coal burned per hour per square foot of grate	pounds,	29.5
Coal per I. H. P. per hour		1.4
Indicated horse-power		8,000

"Six runs were made and the mean results were as follows: —

"Steam pressure 199.5 pounds; air pressure in stokehold, $\frac{3}{16}$ inches; revolutions of port engine, 86.4; revolutions of starboard engine, 86.9; indicated horse-power, port engine, 4,074; ditto of starboard engine, 4,239; total indicated horse-power, 8,313. Vacuum was 27 inches; speed, 15.8 knots. As weight is an interesting element in view of the system of forced draft, it should be stated that the indicated horse-power per ton of engines is 12; per ton of boilers, 12.4; and per ton of machinery, 6. The following is a sample set of data for one day under ordinary working conditions of the boilers [see Table No. 135]."

From the log of a round-trip voyage of S. S. Kensington, Table No. 136 has been compiled, which serves to indicate in a general way the conditions existing and the results obtained. In the calculation of coal consumed per indicated horse-power, there has been included the amount used at sea for the auxiliaries and all other purposes on board the ship. The measurement of power from steam made by the boilers is indicated by that passing through the main engines only.

Table No. 136. — Results from Log of Round-Trip Voyage of S. S. Kensington, Operating under Ellis & Eaves Induced Draft with Sturtevant Fans.

ITEMS.		Voyage No. 30.	Voyage No. 31.
Direction bound		East.	West.
Date of departure		June 30, 1897.	July 24, 1897.
Date of arrival		July 11, 1897.	August 3, 1897.
Time of passage, dock to dock	days, hrs., mins.,	10:13:29	10:4:30
Time of passage, sea	days, hrs., mins.,	10:6:16	9:15:31
Average steam pressure	pounds,	195	193.5
Average fan suction	inches of water,	3.5	3.0
Average vacuum in air reservoirs	inches of water,	0.625	0.5
Average vacuum in ashpits	inches of water,	0.75	0.625
Average indicated horse-power		7,841	6,870
Knots per hour		13.74	14.03
Coal per I. H. P.	pounds,	1.43	1.59
Coal per hour per square foot of grate	pounds,	29.3	28.66
Indicated horse-power per square foot of grate		20.47	17.93
Indicated horse-power per square foot of heating surface		0.643	0.564
Temperature of gases at fan discharge	degrees Fahr.,	427	383
Temperature in air reservoirs	degrees Fahr.,	329	296
Temperature of atmosphere	degrees Fahr.,	62	58
Temperature of fire-room	degrees Fahr.,	83	87

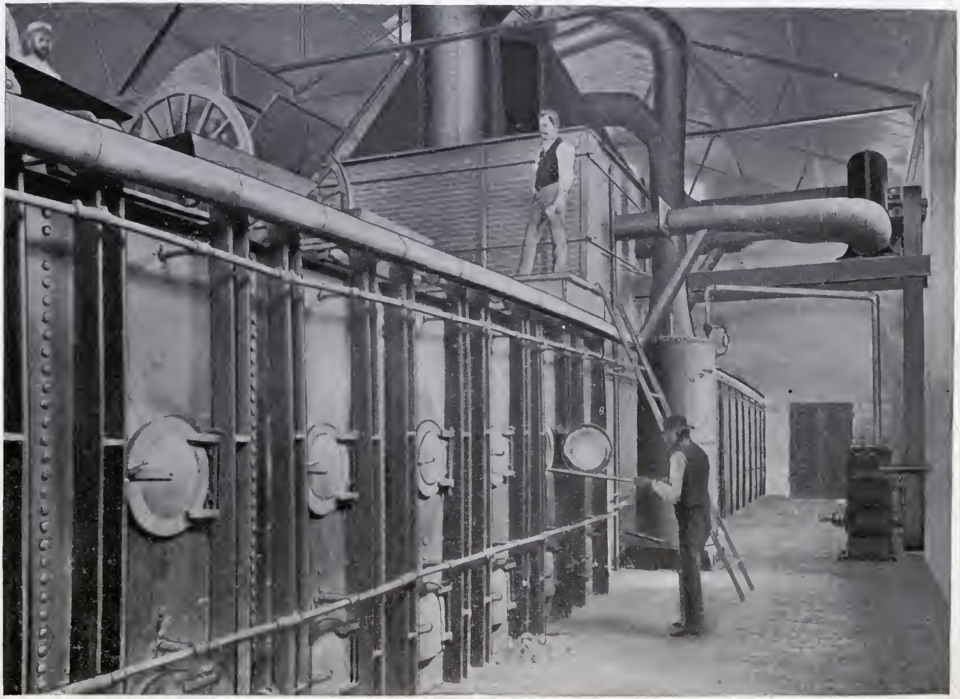

Fig. 94. Washington Garbage Crematory with Sturtevant Blower, at Washington, D. C.

Washington Garbage Crematory, Washington, D. C. — This extensive plant, a view of which is presented in Fig. 94, was constructed by the American Garbage Cremator Company, and equipped with a Sturtevant blower of the "Monogram" pattern placed overhead. This blower is driven by a double enclosed upright engine standing upon the floor and belted upward, as shown in the illustration.

As described,[1] "the garbage crematory is located on the banks of the Eastern branch, directly south of the Capitol. It consists of a series of furnaces heated to a high degree, into which the garbage is dumped directly from the collecting wagons. The heat dries the garbage, and soon incinerates it, making an ash that contains a percentage of fertilizing material which the contractor sells.

"The cremation furnaces, for there are two of them, are the largest reverberatory furnaces ever built in this country. The extreme length of each furnace is 43 feet; width over all, 12 feet; height, 11 feet. The furnaces are located on either side of a self-supporting steel stack, 7 feet in diameter and 120 feet high. Each furnace is fitted up with a Brown patent combustion chamber, which is a combination gas producer, mixer and combustion chamber. In this chamber there is made, and from it is forced, what is known as a powerfully oxidizing flame, competent to destroy garbage in a perfectly sanitary way and at a reasonable cost. The combustion chambers are located at either end of each furnace farthest from the stack, and are connected to the reverberatory burning chambers, over and under which the oxidizing flames are forced, impinging on the upper and lower sides of the garbage at the same time. The garbage itself is deposited on a horizontal double row of semi-steel grate bars, which rest on the sides of the burning chambers on supporting ledges of fire-brick masonry; while their centres are supported by a longitudinal bridge wall of the same material.

"The flames not only pass over and under the garbage, consuming it as they pass, but a portion of the products of combustion passes back through an underneath chamber, or a tritatory chamber, called evaporating cells, where all the water drained from the garbage is deposited. This water is thus evaporated, and in the process of evaporation forced forward, meeting the main or direct volume of flame on its route to the stack. The so-called direct flame is always competent to decompose the evaporated water. However, to make this doubly sure, a stack fire of sufficient intensity is always in operation, over which the entire products of combustion of both furnaces are deflected, before reaching an exit to the outer world through the stack.

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[1] The Evening Star, Washington, D. C., Sept. 12, 1896.

"In the stack is located a series of regenerative tubes, through which air is constantly passing, either by natural or forced draft. This air, when drawn by the natural draft of the furnace stack, goes directly through the regenerative tubes, thence through the hollow longitudinal bridge wall supporting the grate bars, and thence into the combustion chambers, thus supplying them at all times with superheated air. In connection with this regenerative system, and coupled to it, is a powerful [Sturtevant] blower operated by an engine. When the furnaces are fully charged and the final operation in the destruction of the garbage begins, the engine and blower are started, thus giving a more powerful hot blast than could otherwise be obtained."

The general arrangement of the combustion chamber, with the grate bars and garbage thereon, is clearly shown in Fig. 95. This represents one-half of the plant shown in Fig. 94. The air blast from the blower enters the sub-chamber through the pipes which are shown within the stack.

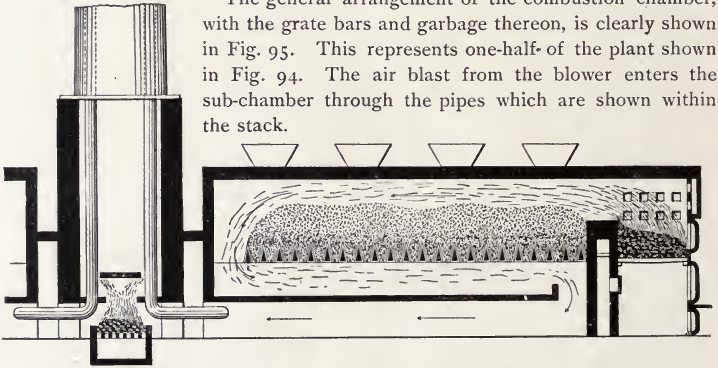

Fig. 95. Section Through Combustion Chamber of Garbage Crematory, at Washington, D. C.

Knickerbocker Lime Company, Mill Lane Station, Pa.— Six single and one double kiln constitute this plant. The apparatus is a 6 x 3½ Sturtevant steel-plate steam fan driven by direct-connected horizontal engine arranged as shown in Fig. 96. The air is discharged vertically upward into a pipe which, turning horizontally at about 18 feet above the floor, passes along the fronts of the kilns. Each single kiln is provided with two grates, one upon either side, each of about 15 feet square feet area, upon which about 4 tons of coal are burned per day of 24 hours. Fig. 97 presents a view of the front of the double kiln and indicates the manner of introduction of air to the ashpits just at the side of the ashpit doors. The lime through which the hot gases pass is contained in a central circular chamber of considerable height, whence it is drawn, when properly burned, through a separate opening.

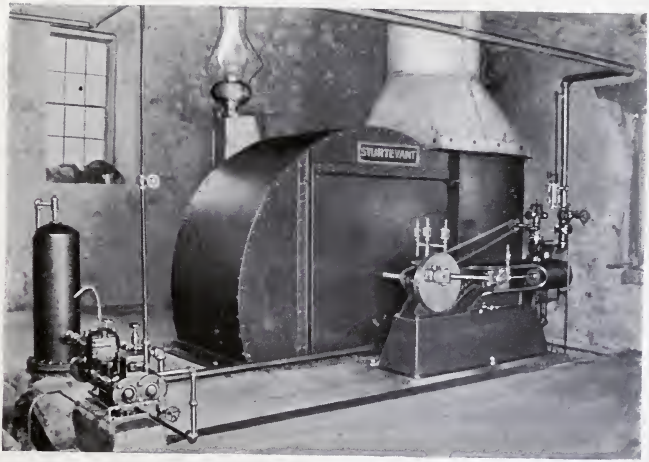

FIG. 96. FORCED-DRAFT APPARATUS, KNICKERBOCKER LIME CO., MILL LANE STATION, PA.

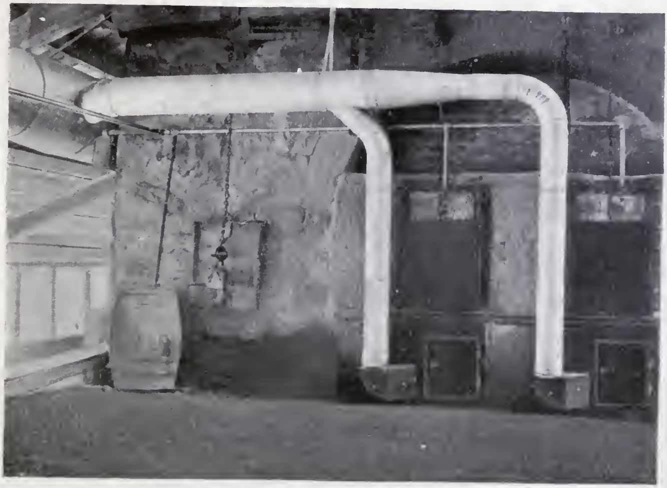

FIG. 97. FRONT OF DOUBLE KILN, KNICKERBOCKER LIME CO., MILL LANE STATION, PA.

Boston Woven Hose & Rubber Company, Cambridgeport, Mass. — The horizontal return tubular boilers of this plant which were tested as here indicated, were equipped with the Bacon setting, which is thus described: [1] "The Bacon setting is a new and useful device, built on strictly scientific principles, having for its objects the saving of fuel and prevention of smoke, or the complete combustion of coal. These objects are most effectually obtained in this device.

"First. The harder a boiler with Bacon setting is worked the better the results, both as regards saving in fuel and the completeness of combustion.

"Second. The admission of air at a very high temperature of heat being equal to that of the gas, and when coal is put on the air exceeds that of the gas. . . .

"Third. Absence of dirt in back connection.

"Fourth. Cleanliness of tubes, which is due to the consumption of gases.

"Fifth. The increase of temperature in flue, which allows shortening of grate.

"Sixth. The holding of full pressure of steam when fires are banked."

Tests were made upon boilers 5 and 6 of this plant by Mr. Greely S. Curtis, Jr., upon Dec. 9 and 11, 1896, [2] "to determine the evaporative efficiency of Dominion coal when burned on the Bacon setting under forced draft. The boilers were fired by the regular fireman, who was experienced in the use of Dominion coal in the settings tested; and they supplied the plant with slightly more than the customary power, as the dynamos were run longer than in other months and the main engine and shafting were driven through the noon hour."

These tests having shown that a great and unnecessary excess of air had been supplied, another test was made on Dec. 23. [3] "The test was to show the effects of lessening the air supply to the boilers tested on Dec. 9 and 11. Since that time, various unintentional air outlets, as well as several of the regular perforations of the Bacon boiler settings, have been stopped up, as was recommended in my previous report.

"The blower was driven at 269.7 revolutions per minute, instead of 350 as previously, and the forced-draft pressure was reduced from 4.9 inches of water to 3.9 inches. . . . The general conditions were similar to those of the previous tests, except that on the 23d the boilers had to supply a slightly greater demand for steam. The following brief comparison may be of interest. [See Table No. 137.]

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[1] The Bacon Setting, Circular. Bacon Engineering Company, Boston, Mass.

[2] Report of Mr. Greely S. Curtis, Jr., Dec. 17, 1896, to The Dominion Coal Co., Boston, Mass.

[3] Report of Mr. Greely S. Curtis, Jr., Dec. 24, 1896, to The Dominion Coal Co., Boston, Mass.

Table No. 137.—Comparison of Results of Tests of Boilers 5 and 6 with Bacon Settings with Sturtevant Fan at Boston Woven Hose & Rubber Company, Cambridgeport, Mass.

ITEMS.		Dec. 9.	Dec. 11.	Dec. 23.
Feed water per pound of coal	pounds,	8.82	8.84	10.27
Equivalent evaporation per pound of combustible from and at 212°, gross,	pounds,	10.63	11.00	12.98
Equivalent evaporation per pound of coal from and at 212°, corrected for priming and steam to blower engine, net,	pounds,	9.56	9.69	11.56

The general results of the second series of tests described above are herewith presented in Table No. 138.

Table No. 138.—Data and Results of Tests of Boilers 5 and 6 with Bacon Settings with Sturtevant Fan at Boston Woven Hose & Rubber Company, Cambridgeport, Mass.

Date		Dec. 23, 1896.
Duration	hours,	11.45
Weight of dry coal consumed	pounds,	7,876.5
Weight of ashes and clinkers	pounds,	631
Weight of water evaporated	pounds,	80912
Average boiler pressure	pounds,	108.52
Average temperature of feed water	degrees Fahr.,	94.78
Forced draft	inches of water,	3.9
Per cent of moisture in coal		8.31
Per cent of moisture in steam		1.70
Equivalent evaporation, dry steam per hour, feed 100°, pressure 70 pounds, inclusive of steam to blower engine,	pounds,	6,887.9
Horse-power developed on A. S. M. E. basis of 30 pounds per horse-power		229.26
Coal consumed per hour per square foot of grate surface	pounds,	10.41
Equivalent evaporation per square foot of heating surface per hour, from and at 212°,	pounds,	4.30
Feed water per pound of coal	pounds,	10.258
Equivalent evaporation per pound of coal from and at 212°	pounds,	11.951
Equivalent evaporation per pound of combustible from and at 212°,	pounds,	12.93
Actual dry steam supplied for useful work per pound of coal	pounds,	9.931

The draft pressure maintained as indicated above is such that a chimney would be absolutely inadequate. It is evident that nothing but mechanical means, as a fan, is capable of producing the required pressure.

Crematory, Lee and Pennsylvania Avenues, Washington, D. C. — The discussion of the relative merits of cremation and burial has naturally resulted in the perfection of certain forms of crematory furnaces for the proper incineration of the deceased. Obviously such a furnace must possess certain features peculiar to itself, among which are rapidity of action and perfect accomplishment of the desired result. In this connection, the fan is of special service in that it renders the operation positive and produces the intensity of draft desirable in rapid combustion. The above-named plant consists of a Brown Incinerator and a Sturtevant Steel Pressure Blower, — the latter, being behind the former, does not appear in Fig. 98.

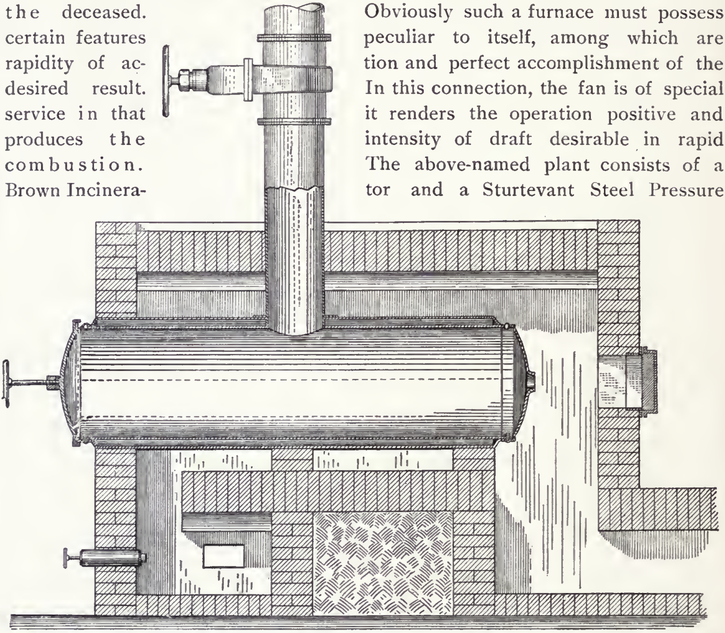

FIG. 98. LONGITUDINAL SECTION THROUGH THE BROWN INCINERATOR IN CONNECTION WITH WHICH THE STURTEVANT FAN IS EMPLOYED.

The general construction and operation of the incinerator is thus described: —

[1]"It requires but half an hour in this incinerator to heat the retort to a proper degree for receiving the body. Within two hours the body is reduced to five or six ounces of pure white ashes, and not one single particle of deleterious gas escapes in any direction; and the cost of cremation is less than a dollar and a half for each time.

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[1] Cremation: Why and How. The Brown Incinerator Company, Boston, Mass.

"The description of the Brown Incinerator is briefly this: It is a structure ten feet long by seven wide, and six high. It is built of fire-bricks, surrounded on four sides by a jacket which is filled with moving water, while the top is covered with a layer of sand. This environment of water and sand keeps the fire-bricks at an equable temperature, so that it does not scale and crack; and, moreover, it so entirely confines the heat within the furnace that, in the midst of cremation, when in the retort there is a temperature of 2,000 degrees, one may lay his hand anywhere upon the outside of the furnace.

"The retort that receives the body is a cylinder eight feet long by thirty inches in diameter, made of decarbonized fire-box steel, which will endure a temperature of nearly 3,000 degrees. In the forward end of this cylinder is a door, which closes absolutely air-tight. Below the door of the cylinder, near the ground, enter three concentric pipes, opening into a small combustion chamber. The innermost of these pipes contains steam, the second pipe crude petroleum, while the third contains burnt gases which have been drawn out from the combustion chamber itself, and are returned to it. These three pipes, emptying their contents at the same point, have this effect: the steam gasifies the oil, and this gas, in turn, mingles with the gases of the outer pipe, forming a new gas of the highest combustibility. This is ignited as it enters the combustion chamber. Then, in a high state of combustion, it rises, and entirely envelops the steel retort in which the body is placed, and circling round it, passes off at the rear of the furnace into a chimney. After combustion has been in progress for half an hour, the retort has reached a heat of 2,000 degrees. The body to be incinerated is wrapped in a prepared sheet, and placed in a semicircular shallow casket. A car conveys this to the mouth of the cylinder. It is then placed within, and the door closed. The intense heat of the retort immediately decomposes the body, liberating its particles in the form of gas. The retort is surrounded by a jacket; from the retort a pipe leads into the jacket, and from this jacket another leads through the walls of the furnace, and connects with a steam-pipe, through which a rush of steam creates, by means of a siphonic construction, a strong suction. This suction is imparted first to the jacket around the retort, and then through its connecting pipe to the retort itself. When the steam is forced through the outer pipe, the gases rising from the body within the retort are drawn first into the surrounding jacket. While passing through this jacket, they are subjected for some time to a very high degree of heat, so that, if they were not already decomposed, they are here entirely resolved into their constituent gases, and absolutely purified. Then they pass out into the steam, and are conveyed away. In this way it is absolutely impossible for a particle of gas to escape from the retort into the surrounding air. A small opening at the foot of the retort permits the entrance of heated air, completing the draft over the burning body. This draft is so strong that even if the door were opened at the expiration of an hour, — after which time the greater part of the gases have been driven away, — there would be no emission of odor or of gas from the retort.

"The rapidity with which this incinerator works, requiring but a half hour for heating and but one hour for the accomplishment of cremation, the very small expense of the fuel, and the inexpensiveness of the construction itself, make it equally serviceable for city cemeteries, where much work is to be done, and for rural cemeteries, where the matter of expense is a great consideration."

The air blast from the Sturtevant blower is introduced through an opening in the farther side of the incinerator, which as already stated does not show in Fig. 98, because of the intervening retort.

The Riordon Paper Mills, Limited, Merriton, Ont. — The adaptability of small Sturtevant fans to the production of draft in place of a chimney is very clearly shown by the experience with this plant. The actual cost of the two fans was only about 6 per cent of the probable cost of the chimney as specified below, to say nothing of the increased convenience, the positive character and the ability to burn cheap fuel. The plant and its operation are thus described: [1] "We have four steam boilers, 150 horse-power = 600 horse-power. They are worked at 90 pounds pressure, and feed water is supplied at an average temperature of 150° Fahr. The quantity of steam used varies considerably as the various machines or digesters in the mill are started or stopped, but during the greater part of the twenty-four hours, the boilers are pushed to their utmost capacity. The boilers are coupled in pairs to two Sturtevant Steel Plate Exhausters, Nos. 50 and 60, both running at 800 revolutions per minute. The draft from the No. 50 is 2-ounce pressure and from the No. 60 2½-ounce pressure. Pennsylvania bituminous run-of-mine coal is used. Consumption is 12 pounds of coal per square foot of grate surface per hour. The fans are driven by a water motor, and require 3½ and 4 horse-power, respectively. They require little or no trouble or attention; the bearings running in water, no lubricant is used. As to the cost of building a stack to fulfil the same requirements, it would be great, probably not less than $3,000, but apart from the question of cost, want of space would make such a structure nearly impossible. The No. 50 fan has now been working over five years with but very little repair, and the No. 60 for two years with no repairs whatever."

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[1] The Riordon Paper Mills, Limited, Merriton, Ont. Letter of February 18, 1896, to B. F. Sturtevant Co.

Boston Duck Company, Bondsville, Mass. — In the case which is herewith presented in Fig. 99, a Sturtevant steel-plate steam fan is used in connection with the Columbia stokers. The fan delivers the air into an underground duct extending along the fronts of the boilers. From this duct special pipes conduct it to the boiler furnaces. The location of the fan and the substantial and unobtrusive form of the special pipes render the arrangement extremely convenient. The general method of operation is evident in the longitudinal section, Fig. 100, but is rendered still clearer by the detailed explanation which here follows:—

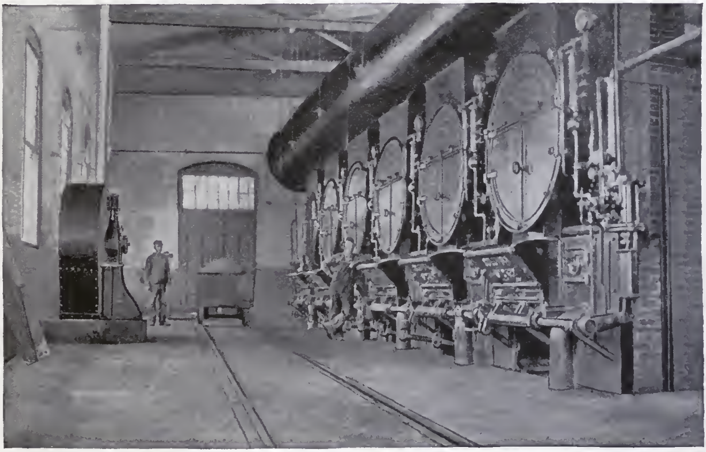

FIG. 99. BOILER PLANT EQUIPPED WITH COLUMBIA STOKERS AND STURTEVANT FAN, AT BOSTON DUCK COMPANY, BONDSVILLE, MASS.

[1] "The fuel to be burned is shovelled into a hopper, the top of which is located about four feet above the floor line; from there the fuel is pushed up through a slightly inclined fuel passage by a slide or coal pusher which derives its motion from an oscillating shaft extending across the front of the stoker; the amount of travel of the coal pusher being regulated by a feed screw, and may be varied from one to four inches, provision being also made to stop the feeding apparatus entirely without interfering with the other stokers which may be set in a battery of boilers and operated by the same main shaft.

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[1] The Columbia Mechanical Stoker and Smokeless Furnace. Catalogue, 8 pp. The Columbia Stoker Company, Holyoke, Mass.

The main shaft receives its oscillating motion direct from a water engine bolted to the stoker front. The coal being pushed through the enclosed fuel passage is delivered on to the blast grates which rest on the air box, which is supplied with air under mild pressure from a fan blower. Owing to the incline in the fuel passage, the fresh fuel will have a tendency to slide along the fuel plate directly on to the blast grates, and, in doing so, cause the bed of incandescent fuel, in course of combustion, to bulge or rise up; the heat from the burning fuel will slowly liberate the gas from the incoming fresh coal, and the air forced through openings in the blast grates in passing up through the fresh fuel will be thoroughly mixed with the gases liberated before passing through the burning fuel above, resulting in a bright, clear fire and the complete consumption of all combustible elements in the fuel. The incoming fuel (on account of being pushed up the incline), is in a compact mass, and will not permit cold air to pass into the furnace, as the whole grates are covered with an even layer of coal. This process of feeding in fresh coal which raises or replaces the ignited fuel is going on continuously, the resulting ash and clinkers being gradually forced over the top on to the inclined grates for final combustion, air being supplied through the openings in the grates induced by ordinary chimney draft.

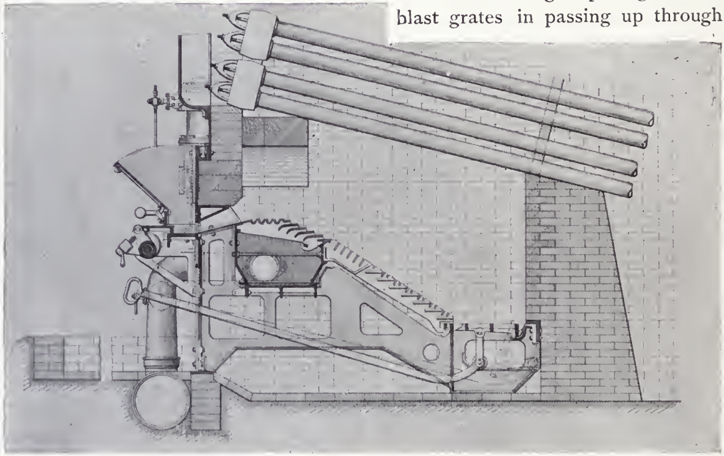

FIG. 100. LONGITUDINAL SECTION THROUGH FURNACE SHOWING CONSTRUCTION OF COLUMBIA MECHANICAL STOKER AND ARRANGEMENT FOR INTRODUCING AIR SUPPLY.

“A drop or dump grate, provided for the removing of clinkers and ashes, is located on the lower end of the inclined grates, and operated by handle bars extended through the front of the furnace. The whole operation of removing the clinkers and ashes only requires a few minutes, and can be done without interfering with the regular course of firing.

“To secure the best results a heavy body of coke should, at all times, be carried in the front of the furnace, and as the amount of coal burned is regulated by the quantity of air supplied, the carrying of a heavy fire during the time when little steam is needed, will not cause loss in fuel, as the air supply to the stokers is regulated by an improved damper regulator.”

The relative efficiency with which this plant operated without and with stokers and forced draft is evidenced in these statements. The tests cover a sufficient period to render the results conclusive.

“[1]The following is an evaporative test of our boilers before the stokers were applied to the same: —

Weeks ending —

Oct. 3, 1886,	water evaporated per pound of coal from and at 212°,	9.9 pounds.
“ 10,	“ “ “ “ “ “ “ “	10.0 “
“ 17,	“ “ “ “ “ “ “ “	10.1 “
“ 24,	“ “ “ “ “ “ “ “	9.9 “
“ 31,	“ “ “ “ “ “ “ “	10.0 “
	Average	9.98 pounds.

“The following figures represent a test made under conditions as above, after the stokers were applied: —

Weeks ending —

Sept. 4, 1897,	water evaporated per pound of coal from and at 212°,	11.7 pounds.
“ 18,	“ “ “ “ “ “ “ “	11.6 “
Oct. 16,	“ “ “ “ “ “ “ “	11.4 “
“ 30,	“ “ “ “ “ “ “ “	11.4 “
Nov. 13,	“ “ “ “ “ “ “ “	11.6 “
	Average	11.54 pounds.

“These figures do not represent selected cases, as we did not make them continuous every week, but made them occasionally. The latter tests we made with coal which I consider to be 8 to 10 per cent inferior in quality, and which cost us 7 per cent less than the coal used in the hand-fired tests. These stokers have been in operation nearly a year and are giving us entire satisfaction.”

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[1] E. G. Childs, Agent, Boston Duck Company. Letter of November 16, 1897, to B. F. Sturtevant Company.

Hotel Iroquois, Buffalo, N. Y.—The direct economic results of the introduction of mechanical draft under the control of a proper system of regulation are very forcibly presented in this plant. The method of introducing the air to the ashpit, without impingement upon the grates, is indicated in Fig. 101.

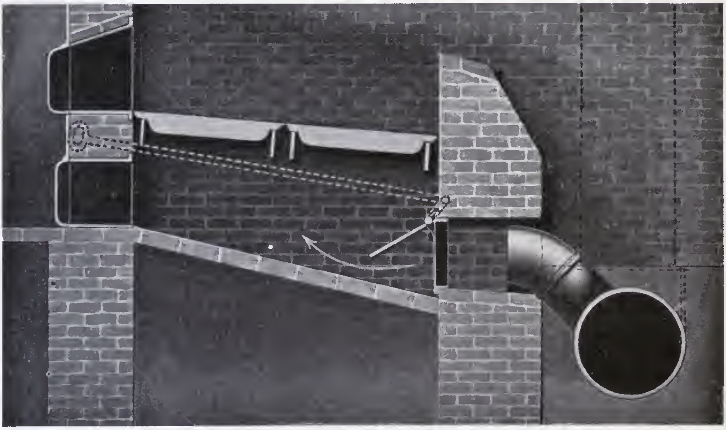

FIG. 101. ARRANGEMENT OF ASHPIT AND DAMPER FOR FORCED DRAFT AT HOTEL IROQUOIS, BUFFALO, N. Y.

[1] "The boiler plant at the Hotel Iroquois consists of four Babcock & Wilcox water-tube boilers, 82 nominal horse-power each, with grates 72 inches long and 46 inches wide, equal to about 23 square feet, the free air space through the gates being 25 per cent of the total area. The fan is a 60-inch Sturtevant, driven by an independent engine whose speed is controlled by the Beckman system of automatic valves, holding the steam pressure within two or three pounds at all times. The Beckman system of automatic valves consists of a pressure-reducing valve, placed in series with a regulating valve, to limit the high speed of the fan, and to set the draft pressure; and a by-pass running around the regulating valve to admit steam to the fan engine, when the regulating valve is closed, thus setting the low speed to keep up combustion and the grates cool. The maximum speed of the fan is 655 revolutions per minute under which an air pressure of about 1¼ inches of water is maintained within the ashpits.

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[1] Wooley & Gerrans, Props. of Hotel Iroquois. Letter of Nov. 17, 1897, to B. F. Sturtevant Co.

"Previous to the installation of this system of mechanical draft and regulation, four boilers were needed to develop the necessary amount of steam, but after its introduction, three boilers proved to be all that were required to accomplish the same results. This reduction in the required size of the boiler plant has been accomplished by a most remarkable decrease in the amount expended for fuel, principally due to the ability to burn a much larger proportion of hard coal screenings. The exact record for two succeeding years is as follows [see Table Nos. 139 and 140, in the last columns of which are given the relative weights and costs which in this case readily serve as a basis of comparison], the average load during the second year being 30 horse-power additional."

Table No. 139. — **Results of Operation of Boiler Plant at Hotel Iroquois, Buffalo, N.Y., without Mechanical Draft.**

Time.	Kind of Coal.	Number of Tons.	Cost per Ton.	Total Cost of Each Kind of Coal	Weight and Total Cost of Coal for Year.
Dec. 1, 1892 to Nov. 30, 1893	Hard coal screenings.	232	$1.25	$1,072.45	4,751.24 tons. $10,157.38
	Hard coal screenings.	601.9	1.30		
	Soft nut.	696.95	2.20	$9,084.92	
	Soft nut.	15.04	2.25		
	Soft nut.	1,759.6	2.30		
	Soft nut.	1,445.75	2.40		

Table No. 140. — **Results of Operation of Boiler Plant at Hotel Iroquois, Buffalo, N.Y., with Sturtevant Fan and Beckman System of Automatic Valves.**

Dec. 1, 1893 to Nov. 30, 1894	Hard coal screenings.	1,299.95	$1.30	$5,356.24	5,013 tons. $7,680.93
	Hard coal screenings.	2,610.8	1.40		
	Hard nut.	3.02	3.50	$2,333.69	
	Soft nut.	843.03	2.10		
	Soft nut.	255.9	2.20		

"These results show an annual fuel saving of $2,467.35, as the result of the introduction of mechanical draft produced by a Sturtevant steam fan, controlled by the Beckman system of automatic valves, a showing which we believe to be amply sufficient evidence of its efficiency."

C. Kiener Fils, Eloyes, France. — An exceedingly convenient arrangement of induced-draft in connection with the boiler plant of a large cotton mill is shown in Fig. 102. The plant consists of seven boilers of special "elephant" type, each provided with a damper in the flue connection, and supplementary dampers so arranged that the gases may pass direct to the chimney or through the fan, or may be first conducted through the economizer and thence direct to the chimney or through the fan as may be desired. The Sturtevant fan which is a No. 100 is driven by a direct-connected electric motor.

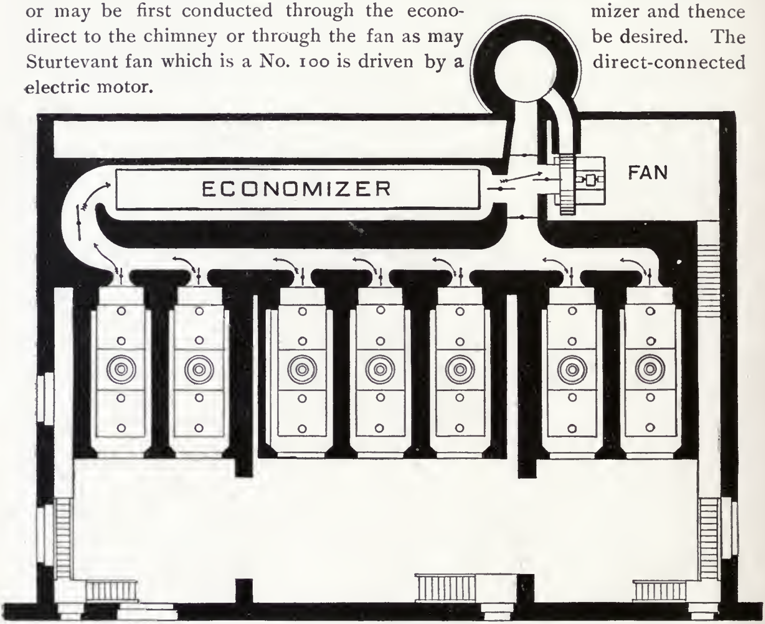

FIG. 102. ARRANGEMENT OF STURTEVANT FAN FOR INDUCED DRAFT AT THE WORKS OF C. KIENER FILS, ELOYES, FRANCE.

"Incognito" Plantation, La. — The arrangement of a Sturtevant blower for burning bagasse is illustrated in the accompanying Fig. 103. [1] "The boilers consist of three batteries of 60-inch x 22 feet return tubular boilers, each having 20 six-inch tubes. . . . The grate surface is made up of specially designed bars, each of which is provided with an individual blast from a main fed by the pressure blower at the left of the boilers. The requisite amount of air for the total number of furnaces is furnished by the blower, and a blast gate regulates the feed to each battery. The mill bagasse carrier feeds the material on a sheet-iron drag, thence on to a drag chain provided with flights, which drags the bagasse along and feeds it into hoppers which are arranged to drop the contents automatically into the furnaces."

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[1] J. H. Murphy, New Orleans, La. Letter of November 17, 1897, to B. F. Sturtevant Co.

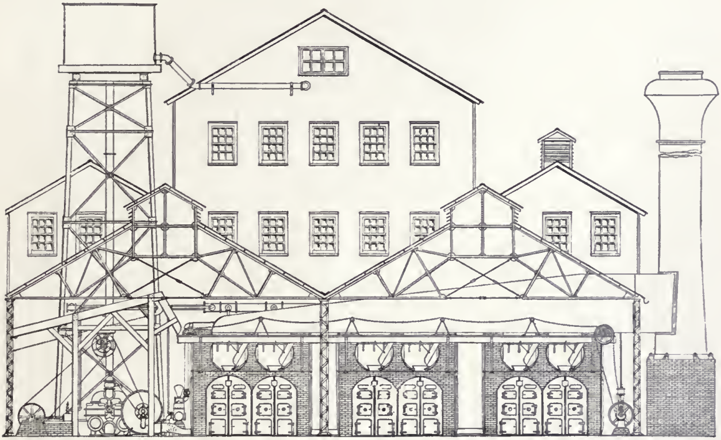

FIG. 103. ARRANGEMENT OF BAGASSE BURNERS WITH STURTEVANT FAN, AT "INCOGNITO" PLANTATION, LA.

Guaranty Building Company, Buffalo, N. Y. — Regarding the operation of this plant by means of a Sturtevant fan, it is stated that:

[1] "(1) Less air is required to produce perfect combustion. For instance, in burning one pound of coal with chimney draft, we require from 24 to 26 pounds of air, whereas with mechanical draft we require only from 18 to 20 pounds of air per pound of coal. This represents a saving of from 10 to 15 per cent in fuel, as we have that many heat units saved by not being obliged to heat this extra quantity of air. (2) A cheaper grade of fuel can be burned, and a finer fuel, because the fan is capable of delivering sufficient air to produce a perfect combustion. (3) With a mechanical draft we can hold back the gases which are lost with chimney draft, by closing the stack damper, and surround every

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[1] Guaranty Building Company, Buffalo, N. Y. Letter of Dec. 3, 1897, to B. F. Sturtevant Co.

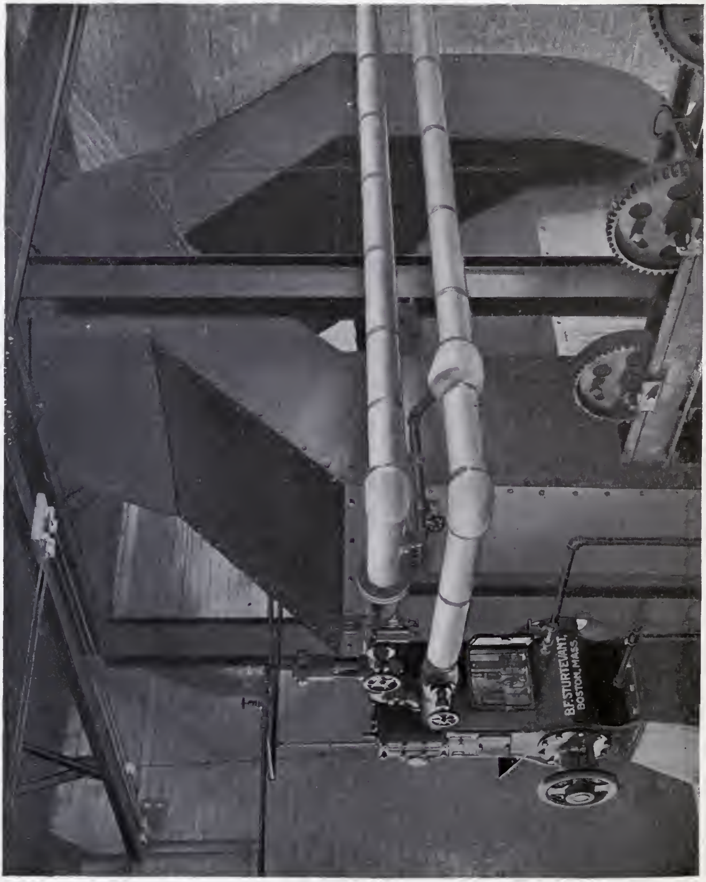
B.F. STURTEVANT.
BOSTON, MASS.

Fig. 104. Induced-Draft Plant at Stamford Gas and Electric Company, Stamford, Conn.

square inch of heating surface with a very high heat. This is especially the case when the boilers are working to their full capacity, or close to it. (4) Chimney draft requires from 10 to 12 square feet of heating surface to produce 1 horse-power, while the same result has been produced by the mechanical draft with from 6 to 8 square feet. (5) A mechanical draft is entirely under the control of the engineer, while the chimney draft is not. The engineer can thus overcome with the mechanical draft all variations in the outside temperature. (6) A mechanical and automatic draft system is especially valuable because it saves all the unnecessary expansion and contraction of the boiler plates, tubes, etc., thus prolonging the life of the boiler. (7) We have no trouble in carrying our steam at the required point needed for our large electric generators, which was a difficult thing to do without mechanical draft. (8) We are also able to avoid all trouble with smoke, although using a cheap grade of fuel. Our combustion is *perfect*. Our plant is a most economical one."

Stamford Gas and Electric Company, Stamford, Conn. — Two special Sturtevant steel-plate steam fans, with full housings, placed side by side with an inlet chamber between, comprise this induced-draft plant. Each fan is driven by a direct-connected double upright enclosed engine, with the wheel upon the end of the extended shaft, the engine bearings being water-cooled. The entire apparatus, as shown in Fig. 104, is placed on the level of the top of the boilers and nearly above the economizer. The gases, led from the back of the boilers, pass through or by-pass the economizer, as may be determined by the position of a damper, and thence enter the chamber between the fans. By means of dampers therein the gases may be caused to enter the inlet of either fan, and thence pass to either of the outlet connections which unite in the stack above. At this junction is another damper. By proper manipulation of dampers either fan may be entirely shut off and rendered accessible through doors which are provided. The speed of the engines is controlled by special regulating valves so as to always produce the requisite draft.

The boiler plant at present consists of four 200-horse-power Wood water-tube boilers set in two batteries. No direct tests have been made, but the owners [1] "have been experimenting with various mixtures of coal, and have found no trouble in getting all the draft we have needed so far, and believe the plant you erected for us will do what you claimed for it." The fans were designed so that either would handle the gases at 300°, after passing through the economizer, and maintain a vacuum sufficient to burn 25 pounds of coal per square foot of grate.

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[1] Stamford Gas and Electric Co., Stamford, Conn. Letter of Oct. 27, 1897, to B. F. Sturtevant Co.

Société Alsacienne de Constructions mécaniques, Belfort, France. — The accompanying Fig. 105 serves to indicate the general arrangement of this plant. It consists of six boilers of the "elephant" type, of which only two are here shown. The fan is a No. 100 Sturtevant, driven by a belted electric motor at about 550 revolutions per minute. The working pressure is 6 kilogrammes per square centimetre, equivalent to about 90 pounds per square inch.

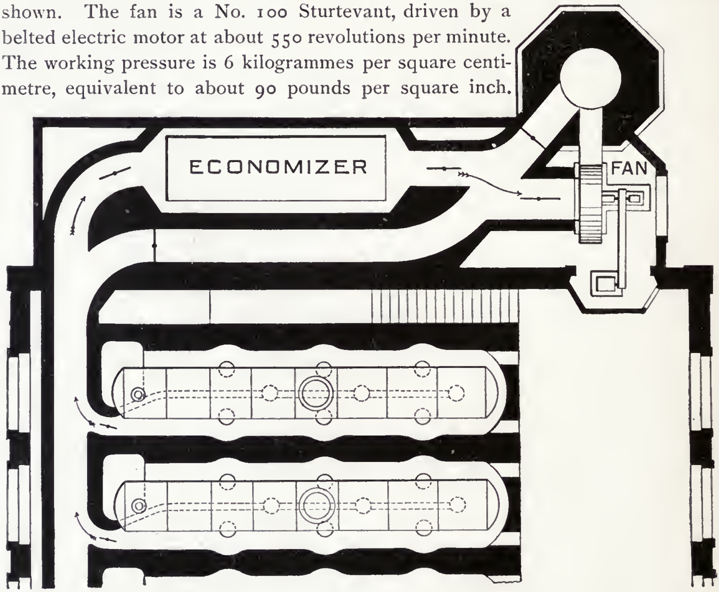

FIG. 105. ARRANGEMENT OF STURTEVANT FAN FOR INDUCED DRAFT AT SOCIÉTÉ ALSACIENNE DE CONSTRUCTIONS MÉCANIQUES, BELFORT, FRANCE.

The general conditions of operation of the plant are as follows: —

Heating surface per boiler	65 square metres.
Maximum production of steam required per hour,	7,000 kilos.
Maximum combustion of coal per hour . .	1,000 kilos.
Total grate surface of the six boilers . . .	14.4 square metres.

Each boiler connects with the general flue by a duct 450 millimetres by 750 millimetres. The general flue has, at its entrance to the economizer and at the chimney, an area of 1.44 square metres. The arrangement of the dampers is such that the gases may be caused to pass by or through the economizer, and thence either through the fan or direct to the chimney.

Leon Godchaux, Elm Hall Plantation, La.—The accompanying illustration, Fig. 106, represents a plant of bagasse burners installed by Haubtman & Loeb. It consists of six cylindrical tubular boilers, each 72 inches diameter by 24 feet long. The necessary air supply for combustion is furnished by a No. 9 "Monogram" Sturtevant blower, driven by an independent engine, and arranged as shown. Each of the boilers is provided with a separate burner of the type known as a "Dutch Oven." The entire burner is surrounded by an air space in the brickwork, with which space it is in communication through numerous small tuyere holes. The air forced into the surrounding space, at a pressure of about 1½ ounces per square inch, is thereby admitted both above and below the fire in the required proportion. Hollow blast bars beneath the grate bars are also employed, so as to thoroughly distribute the air. The bagasse is fed to the furnaces through hoppers, as shown. Each hopper valve is provided with a counter balance, so that the bagasse may be fed with the minimum passage of air through the opening during the process.

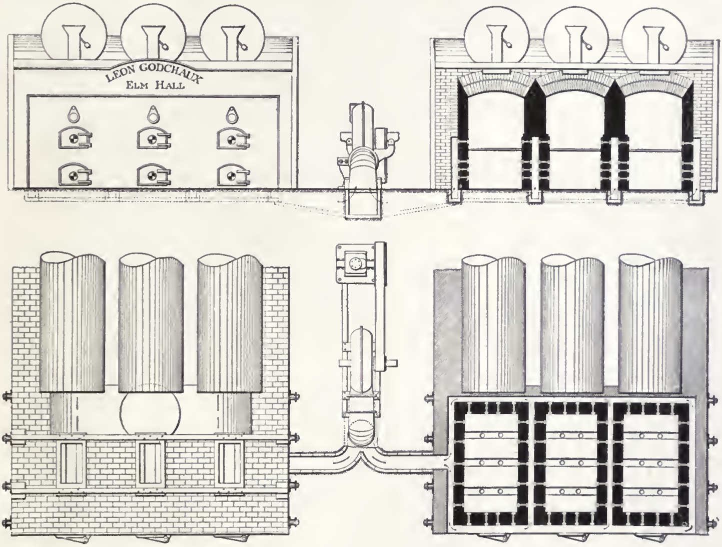

FIG. 106. ARRANGEMENT OF HAUBTMAN & LOEB'S BAGASSE BURNERS WITH STURTEVANT FAN AT ELM HALL PLANTATION, LA.

Otto Colliery, Philadelphia & Reading Coal and Iron Company, Branchdale, Pa. — A general view of this colliery is presented in Fig. 107. The boiler plant, to which reference is here made, is located just over the crest of the ridge, the tops of the three stacks being visible. The following description and report of very carefully conducted tests serve to make clear the somewhat novel arrangement which has been here adopted for economizing fuel even where it is abundant. They most emphatically indicate the necessity of mechanical draft for securing the desired results: —

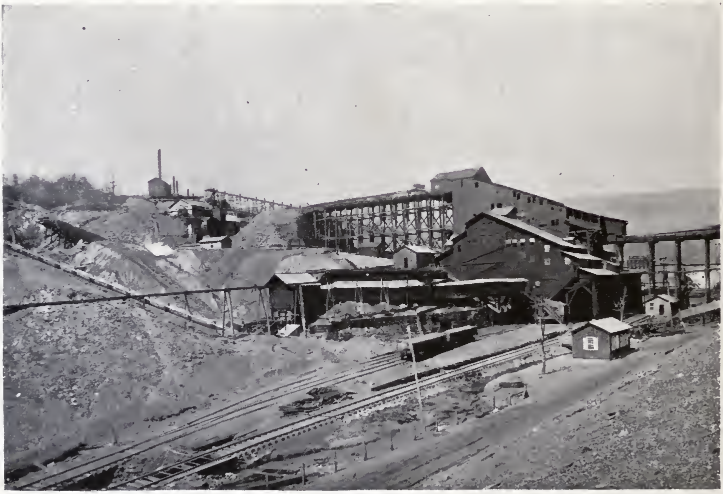

FIG. 107. OTTO COLLIERY, PHILADELPHIA & READING COAL AND IRON COMPANY, BRANCHDALE, PA.

[1] "Anthracite coal is burned under all of the boilers, this coal being refuse termed 'rice mixture,' which is the refuse left after pea and buckwheat are screened out of it. It is the poorest kind of fuel imaginable, and can only be burned by using blowers. From the accompanying report of test the amount of ash in this coal runs up to 35 per cent, and this 'rice mixture' also contains at times 10 per cent of water. The drawing shows the air duct but not the

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[1] Thayer & Company, Boston, Mass. Letter of Nov. 26, 1897, to B. F. Sturtevant Co.

blowers, which are located at the end of the boiler house in a separate building. At the present writing, the three Cahall boilers of 250 horse-power each are set in their respective positions, as shown on the drawing, the first installation proving so satisfactory as to call for duplicates."

The general arrangement of these boilers is indicated in Fig. 108, the Cahall boilers being shown in the rear.

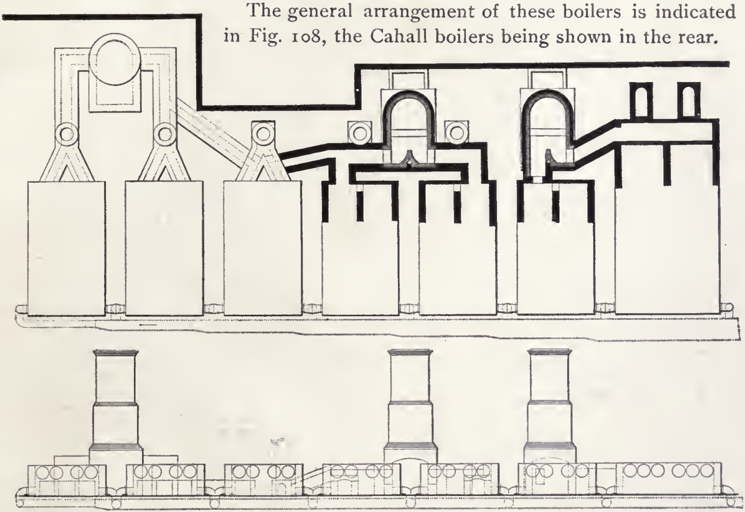

FIG. 108. ARRANGEMENT OF CYLINDER AND CAHALL BOILERS AT OTTO COLLIERY, PHILADELPHIA & READING COAL AND IRON COMPANY, BRANCHDALE, PA.

"In this plant there are 30 plain cylinder boilers, and the Cahall boilers are located between them and the chimney, so as to use the unabsorbed heat before it escapes up the chimney, the gases being forced through the Cahall boilers by means of the blower system, which is absolutely necessary to the accomplishment of the desired results. There is also an auxiliary arrangement for firing the Cahall boilers by coal in the regular furnace in case of derangement or changes in the other part of the plant, so that this same refuse fuel may be burned on the grate in the regular Cahall extension furnace by the blower system. The air blast is carried to the ashpit through the bridge wall of each of the boilers, care being taken that there shall be a large mouthpiece into the ashpit from the air duct, so as to prevent any contracted blowpipe action. The blower system is used throughout all of their collieries. Natural draft could not

Table No. 141.—Results of Tests of Cylinder and Cahall Boilers Under Forced Draft with Sturtevant Fan at Otto Colliery of Philadelphia & Reading Coal and Iron Company, Branchdale, Pa.

Number of Trial		438	439	440	457	458	459
Type of boiler		Cylinder.	Cahall.	8 Cyl. 1 Cah	Cahall.	Cylinder	12Cyl. 1Cah
Number of boilers tested		8	1	9	1	12	13
Duration of trial	hours,	8	8	8	8	8	8
Total water heating surface per boiler	square feet,	988	2,536	3,524	2,536	1,482	4,018
Area of grate	square feet,	126.7		126.7		190	190
Ratio of water heating to grate surface		7.80		27.8		7.80	21.1
Steam pressure in boiler by gauge	pounds,	76.0	76.0	76.0	85.9	85.9	85.9
Force of draft in stack	inches of water,	—0.20	—0.20	—0.20	—0.49	—0.49	—0.49
Force of draft in furnace	inches of water,	+0.07		+0.07		+0.04	+0.04
Force of fan blast	inches of water,	+1.64		+1.64		+1.09	+1.09
Average temperature of feed-water, cyl. boilers,	degrees Fahr.,	141.8		141.8		136.1	136.1
Average temp. of feed-water, Cahall boilers	degrees Fahr.,		60.7	60.7	58.8		58.8
Average temperature of escaping gases	degrees Fahr.,	About 1,600	732	732	711	1,612	711
Total amount of coal consumed	pounds,	21,900		21,900		31,800	31,800
Moisture in coal	per cent,	6.25		6.25		9.5	9.5
Dry coal consumed	pounds,	20,531		20,531		28,779	28,779
Total refuse in coal	pounds,	6,652		6,652		9,785	9,785
Percentage of refuse in coal	per cent,	32.4		32.4		34.0	34.0
Total combustible (dry weight of coal, less refuse),	pounds,	13,879		13,879		18,994	18,994
Quality of steam, dry steam being taken as unity		0.99	1.004		1.002	0.99	
Total weight of water pumped into boiler and apparently evaporated,	pounds,		47,790		76,955	98,200	
Water actually evaporated, corrected for quality of steam, etc.,	pounds,		47,981		77,109	97,212	
Equivalent water evaporated into dry steam from and at 212° Fahr.,	pounds,	77,392	57,289	134,681	92,222	108,496	200,718
Equivalent water evaporated into dry steam from and at 212° Fahr., per hour,	pounds,	9,674	7,161	16,835	11,528	13,562	25,090
Equivalent water evaporated per pound of dry coal from and at 212° Fahr.,	pounds,	3.77		6.56		3.77	
Equivalent water evaporated per pound combustible from and at 212° Fahr.,	pounds,	5.88		9.71		5.71	
Dry coal burned per sq. ft. grate surface per hour	pounds,	20.2		20.2		19.0	19.0
H. P. on a basis of 30 lbs. of water evap. per hr. from temp. of 100° F. into dry steam at 70 lbs. gauge,	H. P.,	280.4	207.6	488	334.1	393.1	727.2
Horse-power, builders' rating	H. P.,	141	250	391	250	212	462
Per cent developed above or below rating	per cent,	99 above	16.9 below	24.8 above.	33.6 above	85.4 above	57.4 above

have allowed this method of utilizing waste heat. By the use of this blower system they secured an increase of steam power capacity amounting to 85 per cent without any increase in the amount of fuel burned. The gain in fuel economy by the use of the Cahall boiler with this system was 74 per cent, that is taking that much out of the gases before they escaped. The use of this blower arrangement not only made this saving and this immense increase in the capacity, but it also showed this without additional labor, which in this single plant as shown on this drawing of 750 horse-power of Cahall boilers, showed a saving of labor and mule feed of $3,822 annually. The mule feed referred to consists in the expense of handling coal. This plant runs 24 hours per day."

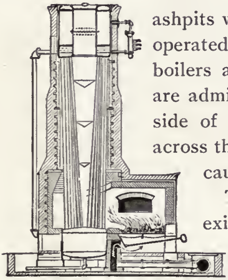

FIG. 109.

The arrangement of dampers for admission of air to the ashpits when required, is shown in Fig. 109, the damper being operated from the front by means of a lever. When the cylinder boilers are discharging the gases into the Cahall boiler they are admitted through the special opening here shown in the side of the combustion chamber, whence they pass upward across the water tubes in the usual manner. The boiler, because of its vertical form, becomes in effect a chimney.

The general specifications of the plant, the conditions existing and the results obtained by careful trials are clearly shown in the following extracts from the official report. The general results are presented collectively in Table No. 141.

[1] "Cahall Boiler: 108 tubes 4 inches by 18 feet; mud drum 68 inches diameter by 48 inches; top drum 80x80 inches; smoke flue in top drum 34 inches diameter by 80 inches.

"Heating surface exposed to water, 2,536 square feet. Steam heating surface, 50 square feet.

"Cylinder Boilers: These are arranged four in a battery and two to a furnace. The shells are 34 inches by 30 feet, set to expose 123.5 square feet of water-heating surface per boiler. The grates are 5 feet long and provide 15.8 square feet to each boiler.

"Stacks: Each battery of four boilers is provided with a stack 34 inches diameter by 36, 51 and 55 feet high, respectively. These stacks were not in use on the tests hereafter described.

"The Cahall boiler has a stack 38 inches in diameter which was 53 feet 6 inches high, above the grates, on trials 438, 439 and 440, and 78 feet 6 inches high on trials 457, 458 and 459.

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[1] Report of Mr. Jay M. Whitham, on Trials of Cahall Boiler using Waste Heat from Cylinder Boiler, 1896.

“Fan Blast: The draft was supplied by a fan blast under the grate, which, on the trials noted, supplied air for the 30 cylinder boilers at this colliery.

“Results of the Trials: Trial 438, shown on annexed trial sheet, was made on 8 cylinder boilers.

“Trial 458 was made on 12 cylinder boilers. On these trials the chief items of interest are: Temperature of waste gases about 1,600° Fahr. Horse-power developed per cylinder boiler, 33 to 35. Evaporation, 212° Fahr., per pound dry coal, 3.77 pounds.

“Trials 439 and 457 relate to the horse-power developed by the Cahall boiler when receiving waste gases from the cylinder boilers. The trials of the cylinder boilers and the Cahall boilers were made simultaneously.

“Trial 439 shows the duty of the Cahall boiler when coupled to 8 cylinder boilers, and trial 457 when coupled to 12 boilers.

“The chief items of interest are as follows: —

	No. 439.	No. 457.
No. of cylinder boilers supplying waste heat .	8	12
Temperature of gases entering Cahall setting .	1,600° Fahr.	1,612° Fahr.
Temperature of Cahall stack	732° Fahr.	711° Fahr.
Horse-power developed by cylinder boilers . .	280.4	393.1
Horse-power developed by Cahall boiler . .	207.6	334.1
Total horse-power developed . .	488.0	727.2
Gain in capacity by use of Cahall boiler, without change in labor employed or fuel burned .	74 per cent.	85 per cent.

“The above results are shown more fully in trials 440 and 459, which represent the combination of cylinder and Cahall boilers. The boiler horse-power developed from hour to hour was as follows for the Cahall boiler, when supplied with gases from 8 cylinder boilers, viz.: —

Hour.				
1st.	Cahall boiler made .	267.0	horse-power,	fires clean.
2d.	“ “ “ .	255.0	“ “	
3d.	“ “ “ .	141.8	“ “	cleaning.
4th.	“ “ “ .	257.0	“ “	fires clean.
5th.	“ “ “ .	224.8	“ “	
6th.	“ “ “ .	175.0	“ “	
7th.	“ “ “ .	183.3	“ “	
8th.	“ “ “ .	156.9	“ “	cleaning.
	Average for 8 hours,	207.6	horse-power.	

“When supplied with waste heat from 12 cylinder boilers, the Cahall boiler developed power as follows:—

Hour.				
1st.	Cahall boiler made .	367.4	horse-power.	
2d.	“ “ “ .	300.4	“ “	cleaning.
3d.	“ “ “ .	384.4	“ “	
4th.	“ “ “ .	317.3	“ “	cleaning.
5th.	“ “ “ .	351.1	“ “	
6th.	“ “ “ .	283.9	“ “	cleaning.
7th.	“ “ “ .	334.0	“ “	
8th.	“ “ “ .	335.0	“ “	
	Average for 8 hours,	334.1	horse-power.	

“Summary: 1. The cylinder boilers are run to develop from 33 to 35 horse-power.

“2. The cylinder boilers by themselves evaporate about 3.77 pounds of water per pound of dry coal.

“3. The combination of cylinder boilers and Cahall boilers, the latter using waste heat only, permits an evaporation of 6.98 pounds of water per pound of dry coal.

“4. The waste gases enter the Cahall setting about 1,600° Fahr., and leave it about 700°.

“5. The use of waste gases by the Cahall boiler increases the available horse-power of the plant from 74 to 85 per cent, according to the number of boilers used for supplying the waste heat.

“6. The 250 horse-power Cahall boiler using waste gases from 8 cylinder boilers developed 207.6 boiler horse-power, and when supplied by 12 boilers it developed 334.1 horse-power, or 33.6 per cent above its rating.

“7. The fuel used, called a “rice mixture,” consisted of 20 per cent slate pickings, 8 per cent buckwheat, 46 per cent rice coal and 26 per cent dirt. It contains, as used at this colliery, from 6.25 to 9.5 per cent moisture, and from 32.4 to 34 per cent ash and refuse.

“It is burned with a strong fan blast.”

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